SEP 1 1926

MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

Trained Men in American Industries

If American industries are to overcome the difficulties which confront them they must begin by overcoming the difficulties which confront the people employed in their shops. As long as countless thousands of partly trained, dissatisfied workingmen drift about the country from one locality to another there will be labor troubles and, more than that, there will be business depressions, because in the long run, the manufacturer's customer is his employee. If American industries are to have a steadily widening market for their wares, they must build up a great class of stable, prosperous employees, well educated and thoroughly trained for their work. Apprenticeship will bring this about.

(Extract from Harold S. Falk's paper on The District Apprenticeship System, the leading article in this issue, to be presented at the New Haven Machine Tool Exhibit on Thursday, September 9, 1926)

SEPTEMBER 1926

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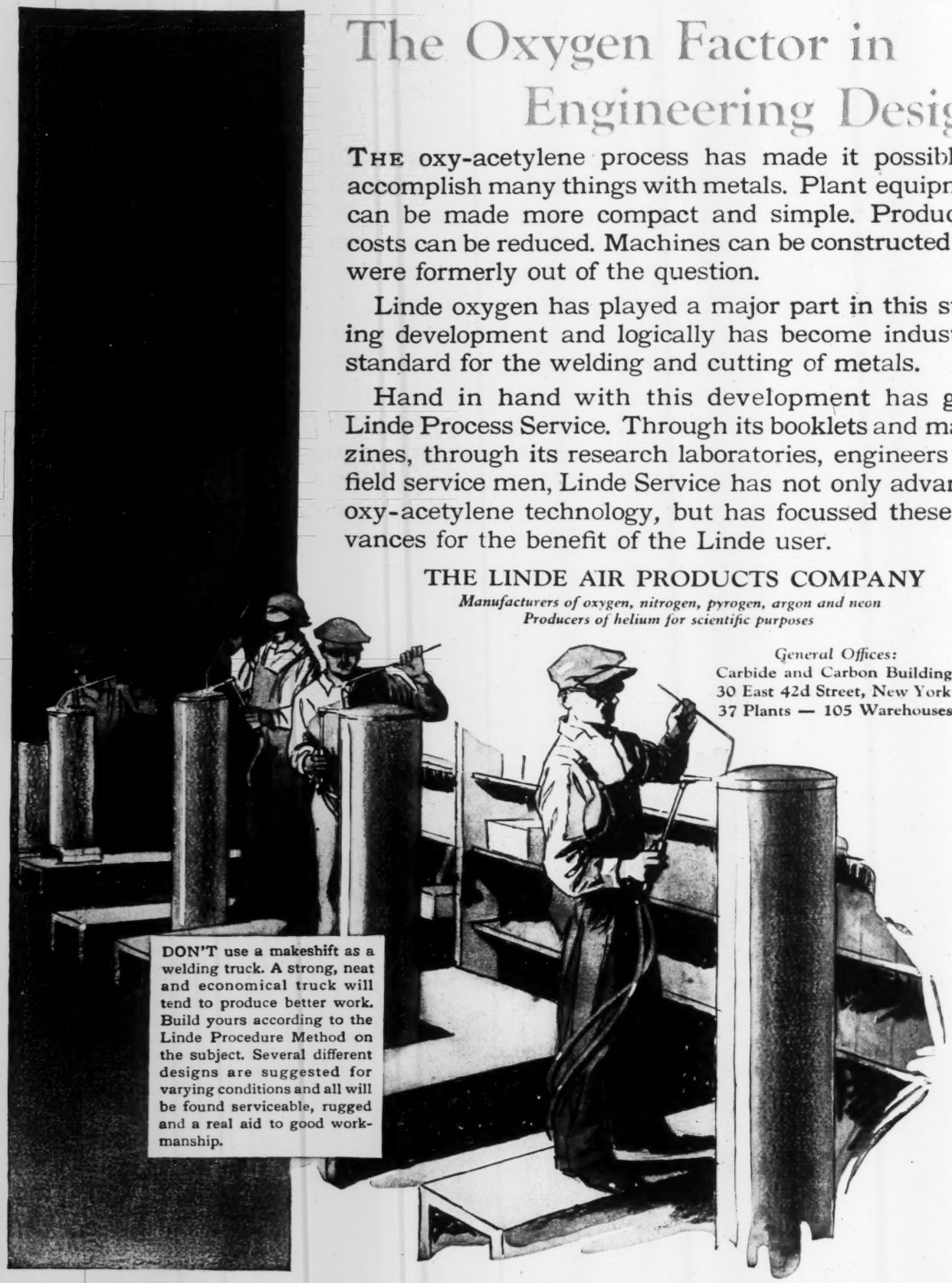

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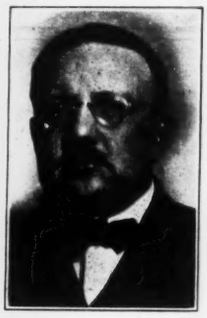
W. H. Clapp

F. H. Penney

C. L'H. Ruggles

H. S. Falk



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MECHANICAL ENGINEERING

Volume 48 September, 1926 No. 9

The District Apprenticeship System

A Successful Coöperative System of Training for Communities of Large and Small Shops—Each Shop Bears Its Share of Responsibility

By HAROLD S. FALK,[1] MILWAUKEE, WIS.

SOME time ago a group of visitors came to see us at Milwaukee. They were a committee representing the manufacturers of a medium-sized but well-known manufacturing city which produces machinery of various kinds. They had heard that the metal trades industry of Milwaukee was conducting a little experiment in district apprenticeship organization and desired to study the methods in use. We expressed some surprise that they should come to us for apprenticeship information as they had been training mechanics successfully for years in their city, in fact, we knew a number of excellent workmen who had learned a trade in one of their shops.

The spokesman of this group of men told the history of their apprenticeship work over the luncheon table and explained why they had come. In their city there is one large plant employing several thousand men and a large number of much smaller shops largely engaged in specialty work. The large plant is an old institution, probably a hundred years old or more, and, up to ten years ago, had been very active in training apprentices. In fact, it is not generally known that they no longer train apprentices. For many years this large plant had been producing mechanics not only for itself but for all the other shops in the city and district. Everything went well enough. Most of the apprentice graduates left this large plant when their courses were completed to work in the other shops. However, enough remained to satisfy all requirements, and as long as the officers of the corporation were willing to train mechanics for the entire district, everyone was satisfied, especially the smaller corporations who were being supplied with skilled mechanics in whose training they had not the slightest part. It was a peaceful, established industrial community and was likely to remain so as long as the large corporation was willing to play Santa Claus in the district.

But ten years ago the large establishment was reorganized. New directors and new officers brought new ideas into the business and every department was gone over very thoroughly to determine where money could be saved. The new officials had been brought up in the financial rather than in the manufacturing field, and when they began their investigation of the apprentice department the apprentice supervisor was called in to justify his presence and work. They pointed out to him that his department was costing the corporation considerably over twenty thousand dollars per year. He admitted that the figure was about right. They asked him how many apprentices he had in the shop. He said about three hundred. They then asked what proportion of those who completed their time remained at work or returned after a year or so in other plants. He answered about a third, becoming very red in the face and uncomfortable.

"And where did the other two thirds go?"

"They went to the other shops around here."

"Do you mean to say that this corporation has been spending nearly twenty thousand dollars per year in training young men for every little shop in town?"

"I'm afraid that's what has been going on."

"That is a crime. Other plants of this size get along without apprentice training. Where do they get their mechanics?"

"Why, they go out and hire them wherever they can. If there are none available in the neighborhood, they import them."

"Well, that is what we shall do from now on. Do not hire any more apprentices."

Things went along well enough for a number of years. Those apprentices who had begun their courses were permitted to finish them, after which they went out to work in the shops of the district as well as in the plant in which they had been trained exactly as they had in previous years.

What Happened When a Large Concern Discontinued Apprentice Training

Then there came a great change. It came gradually, almost imperceptibly, but it came. The last apprentice had been given his certificate and no more were forthcoming. A number of years passed and many skilled workmen had died and others had moved to other districts. There were none to take their places and a shortage gradually developed. The different shops began to bid against each other for the services of those mechanics who remained. After a few more years the rumor of higher wages in the city began to attract transient workers. When business was very good and there were not enough transients available, carloads of workmen were imported from wherever they could be obtained. Labor turnover more than doubled in all plants of the city, and manufacturing costs mounted higher and higher. With the transient workmen came agitators and other undesirables. Labor troubles developed here and there, something which had never been heard of in that city. Portions of the town became disreputable and crimes of various kinds became common. Things went from bad to worse and, while conditions may not have been more serious than in other industrial centers, they were bad enough to bring about a series of joint meetings between various civic organizations and the manufacturers of the city in order to devise some way of restoring the former healthy condition of the community with its orderly population of satisfied, highly skilled workmen.

After much discussion it was decided to reëstablish apprenticeship and, as one speaker expressed it, "get our mechanics from our own schools and not from the blacklists of other towns." However, the officials of the big corporation refused to consider the reëstablishment of any system, no matter how much it was needed, whereby the entire burden of training skilled men for the city would rest upon them. On the other hand, the small shops maintained that they could not possibly train apprentices because their work was too specialized to provide all-around trade training. It was agreed that both attitudes were reasonable, and a committee was appointed to investigate the possibility of some scheme of coöperative district apprentice training.

It was this committee which came to Milwaukee.

District Apprentice Training the Solution for a Congeries of Large and Small Shops

A district apprenticeship program will very likely solve the problems of this industrial center. There is no reason why the large plant should bear the entire burden of training apprentices for the

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Contributed by the A.S.M.E. Committee on Education and Training for the Industries for presentation at the New Haven Machine Tool Exhibition, Mason Laboratory, Yale University, New Haven, Conn., September 7–10, 1926

district, nor is there any reason why the small shops should be excused from taking a proper part in the task of training apprentices, provided the manufacturers of that city are willing to organize for apprentice training, as they probably have organized for other purposes, and coöperate intelligently. The burden of training mechanics can be properly distributed by an interchange of apprentices, justly administered by a central organization. The large shop and the smaller shops should all hire apprentices in the proportion of their employment of skilled mechanics and in such numbers as will insure sufficient new men in the district every year. What this number must be can be calculated from the requirements of former years, labor-turnover percentages, and other sources. Apprentices from the small shop can be taken into the big plant for a sufficient length of time to train them in those departments of the trades which they cannot learn in the specialty shops. On the other hand, the apprentices from the big shop can be sent into the smaller shops for special training in those lines in which the small shops specialize.

On the face of it, this appears to be a simple solution of the problem, but the work will need to be very thoroughly organized in order to overcome many difficulties which will arise. The plan will not work if one shop pays higher wages than another; neither will it work if apprentices from outside shops are not treated with the same consideration in every plant as those who are employed there; nor will it succeed if there is any jealousy over the graduates who may be claimed by various corporations in whose plants they may have worked at various times.

The Central Committee and Its Functions

The first requisite for overcoming these difficulties is a central committee for apprentice training that is capable, impartial, and respected. This may be a committee of the Chamber of Commerce or of a manufacturers' association or trade organization—it makes little difference as long as it has the support of all in the district and as long as its prestige is secure. This committee should establish approximately uniform work schedules and uniform pay schedules for apprentices of various trades and for boys of different ages and educational training. Thus the college-graduate apprentice will be paid more than the high-school-graduate apprentice, and the high-school graduate more than the apprentice who has had only an eighth-grade education. Other matters will need to be determined by this committee. How much time shall the apprentice spend at school and what his compensation for attendance? Shall this school work be done in the public schools, in trade schools, or vocational schools, or shall each corporation establish its own classes? Shall the school work be done during certain hours each week, or shall the apprentice devote a number of weeks per year to continuous school attendance? Shall apprentices be given credit for shop work previously done and how much? All these problems and countless others will come before this committee. Their administration of apprenticeship in a community will be largely responsible for its success or failure. If their decisions and regulations are not upheld by all the shops in the district, the committee will need to be reorganized. As the work progresses, a district apprenticeship supervisor who reports to the committee and receives instructions from it may be engaged, and will soon justify his salary.

An important function of this committee will be to group the plants in a district into a number of self-supporting units as far as apprenticeship is concerned. Small shops may be grouped about a large shop in such a manner that every kind of trade training is available in the group, so that apprentices need not be exchanged except within this group. A number of small shops may be able to provide complete trade training without coöperation with any larger plant. In such a case these small shops may be grouped together for apprenticeship purposes.

In every way the apprenticeship committee will be the governing body for the apprentice movement in the district, and will be responsible for the promotion of apprenticeship in the district as well as for any decisions regarding apprenticeship which may be required.

A district apprenticeship system is necessary in order to distribute equally the burden of apprentice training in a community, but it is still more necessary in order to impress the apprenticeship idea upon the population. Apprenticeship appears to be a phase of shop management, a matter of work schedules and transfer from one kind of work to another and of hiring—and sometimes firing—boys of school age. It is very much more a matter of contacts in the community. Parents and teachers, and pastors, employment agencies, and newspaper men must be told about apprenticeship as well as foremen, superintendents, and the boys themselves. Apprenticeship is not only a matter of training boys but of getting boys of the right caliber. Every one knows that the average American young man is not much disposed to go into industrial work, nor can it be expected under modern conditions. The American ideal seems to be the financial and commercial genius, if we are to judge from our publications, theaters, books, and other agencies which influence the public mind. Read a novel or short story or attend a motion-picture performance and you will invariably find the hero a man engaged in purely commercial work. Every advertisement directed toward the American man shows him seated at a desk in an office. The workingman is never idealized, in fact, it is not so many years since teachers threatened lazy pupils by telling them that they would never be more than mere workingmen if they did not pay more attention to their school work. Parents, however humble in life, will insist that their sons become professional men because sons of friends of theirs have become professional men and nothing less will do for their own children. Is it any wonder, in the face of all this, that American young men do not ordinarily go into the trades unless they are forced into them by circumstances?

Trades Must Be Given a New Dignity and Made More Attractive

The trades must be given a new dignity. They must be made more attractive. The personal requirements for most trades are as high as for ordinary commercial work. The machinist is as good a man as the bookkeeper and he gets as much pay. It still remains to make him as highly respected in the community as the bookkeeper is. Boys will not go into the trades in sufficient numbers until they, and especially their parents, feel that the trade work is just as desirable as clerical work; until the molder and the patternmaker are just as important men in civic organizations and clubs—and in the eyes of the young ladies too—as the bank clerk and the insurance-office man. Apprenticeship will achieve this result if it is made a district undertaking.

One or two isolated establishments cannot affect an entire community, no matter how well established their apprentice work may be. Their influence will be negligible. All the manufacturers in a district must unite in one great organization to advertise the trades and apprentice training for the trades throughout the entire city. People in all classes of life must be taught to take apprenticeship for granted. Apprenticeship must be so thoroughly established that every boy, from his earliest days, will think no more of escaping an apprentice course than of escaping attendance at school. The district apprenticeship committee must confer with school authorities, teachers' associations, and parent teachers' associations. They must establish contact with Boy Scout officials and other organizations for boys and young men. They must go into the schools with apprenticeship exhibits and talks; they must arrange for visits by high-school and grade-school classes to shops and factories; they must influence the school boards in order that more and more of civic studies and English composition may be devoted to industrial questions. They must cause articles to be written for the newspapers and other publications regarding industrial and trade work and the necessity of training for them by means of organized shop courses. With the help of every single manufacturer in the district they must work enthusiastically and unceasingly, year after year, in the face of difficulties of every kind, until the mechanics required in the district are raised, educated, and given a thorough apprentice training in the homes, schools, and shops of the district.

Successful Apprentice-Training Work of Milwaukee Metal-Trades Industries

All this is not mere theory, it has been proved by actual experience. The metal-trades industries of Milwaukee have been engaged in a district apprenticeship program for a number of years and, while a mere beginning has been made, the results justify the most extravagant predictions for the continued and increasing suc-

cess of apprenticeship training as a result of the district organization. There are forty-nine members of the Milwaukee Metal Trades Association and in twenty-nine of these plants apprentices are employed, although only eight shops employ more than twenty-five apprentices. There is an apprenticeship committee of the Milwaukee branch of the Metal Trades Association, and work and pay schedules have been established and are in force. The people of Milwaukee understand that the metal-trades shops offer a real opportunity to young men, and boys of all classes are taking courses in the various shops. High-school and college graduates apply voluntarily for apprentice training even in the foundry, and the teachers and school authorities are so imbued with the apprenticeship idea that the opportunity to undertake an apprentice course is held out to the best students under their instruction rather than considered a method of gracefully dismissing those whose school work is below par.

In achieving this result the Industrial Commission of Wisconsin, which administers the apprenticeship law of the state, and the Milwaukee Vocational School have been of invaluable assistance. The latter institution, with a weekly attendance of about fifteen thousand young people, has been the center of the apprenticeship movement and activity of the city. There, more than anywhere else probably, have the young people of Milwaukee become convinced of the necessity of organized experience as well as organized education. A strong educational institution which can appreciate the attitude of the apprentice and give him related trade instruction under the best possible conditions it can set up will be more effective than any other agency in building up a district apprenticeship system.

From the organization of a number of district apprenticeship systems to the establishment of a national apprenticeship program is a natural step. Our American industries require a certain number of mechanics and other skilled men every year. Just as it is unfair for one or two shops to train all the mechanics for a district, so is it unfair for one or two or even ten industrial districts to produce all the expert men required by the entire nation. And just as a strong district committee is necessary to make apprenticeship successful in a community, so is a national apprenticeship governing body necessary for the establishment of apprenticeship throughout the nation.

Apprentice Training Needed to Build up a Great Body of Stable, Well-Satisfied, and Prosperous Employees

If American industries are to overcome the difficulties which confront them they must begin by overcoming the difficulties which confront the people employed in their shops. As long as countless thousands of partly trained, dissatisfied workingmen are permitted to drift about the country from one locality to another there will be labor troubles and, more than that, there will be business depressions, because in the long run the manufacturer's customer is his employee. If American industries are to have a steadily widening market for their wares they must build up a great class of stable, well-satisfied, and prosperous employees—men who are well educated, thoroughly trained for their work, who settle down, establish homes of their own, and bring new generations into the world with the same characteristics.

Apprenticeship will bring this about. Apprenticeship destroys unrest and dissatisfaction because it creates an interest in work. Moreover, apprenticeship established throughout the country will bring about a stability of population in all centers because each district will produce its own mechanics and thus largely destroy the temptation to drift from place to place. This is borne out by the experience of those cities which have a practical corner on some one industry. Thus, Grand Rapids, Michigan, has a comparatively stable population because there are not many places in which large numbers of furniture workers can find employment, nor are there many places from which large numbers of furniture workers can leave for Grand Rapids.

Manufacturers and business men throughout the country have organized for many purposes. They have organized for advertising, for legislation, for technical research, and for sales. Why should they not organize for the training of sufficient apprentices for every American community? National organization, as the author has remarked on other occasions, has made "Safety First" an American watchword and has saved thousands of lives. Why cannot national organization make another watchword of "Apprenticeship" and make tens of thousands of mechanics?

Many organizations had been interested in the safety movement and had done excellent work in promoting safety, but no important results were obtained until there was organized the National Safety Council, which became the one great American institution devoted exclusively to the prevention of accidents. The National Safety Council took over the safety work of all other agencies and organizations and has done marvelous work in all parts of the country.

A National Apprentice Training Committee Needed

At present there are many organizations which are interested and active in the apprentice movement and all are doing their share. The American Society of Mechanical Engineers, the National Founders' Association, the American Founders' Association, the National Metal Trades Association, a great many organizations in the building trades and many others which do not come to mind at the moment have national apprenticeship committees which are very active. The Chamber of Commerce of the United States has turned its attention to the problem of apprenticeship. This indicates that leaders in all lines appreciate the necessity of training young men for the various industries. But is there not some duplication of effort in all these many enterprises? The corporation which is interested in apprentice training finds itself receiving information from and contributing information and money to a great many different organizations, all of which are promoting apprenticeship. Why cannot these various organizations turn their apprentice work over to one central committee on apprenticeship, similar to the National Safety Council, which shall direct all apprentice work for all these organizations? This central committee would send its own experts into all parts of the country to promote apprenticeship and would represent all organizations which might take a part in forming the central apprenticeship governing body. Until such a step as the formation of such a committee is taken, it is not likely that the apprentice movement will become the great national activity which it needs to become if our industrial life is to become properly stabilized.

More Steel Per Man

IT IS fully recognized that in the manufacturing industries improvements are continually being made in equipment and methods, so that productivity per man is, or rather should be, increased. The human element is a factor. Given the same methods and equipment, men may become more skillful and keen, or less. Improvements supplied by the employer may contribute in part to the workman's having an easier time.

Nearly all that we have now, apart from food, is due to improvements made in approximately a century. One might say that housing and clothing should be excluded in this statement along with food; but the conveniences of our housing today and the multiplicity of articles of clothing worn in the course of a year are such that the amount of housing and clothing of a century ago is but a small fraction of today's.

There has grown up in the last few years a disposition to doubt whether we are now making similar progress. In a long range view the progress is plainly seen without definite measurement. To get an idea of the recent or present pace, precise measurement is requisite. This is furnished for several industries by some studies of Ethelbert Stewart, Commissioner of the Bureau of Statistics. The analysis of labor productivity in the iron and steel industry is made for the years 1914 to 1925 inclusive, with the omission of 1915. Allowance is made for the shift from the 12-hour to the 8-your day whereby the showing is of man-hours rather than of the number of men in employment.

The bare figures appear to show an increase of 50 per cent from 1914 to 1925, but Mr. Stewart warns against acceptance of this as a general conclusion, saying that "this is undoubtedly an exaggeration of actual facts, for the year 1914 was one of severe depression, while 1925 was at least a good year." (*Iron Age*, vol. 118, no. 5, July 29, 1926, p. 301.)

An Experimental Investigation of Hydraulic Air Compression[1]

By PATRICK FOSTER MUMFORD, STANFORD UNIVERSITY, CAL.

THE hydraulic air compressor is the only practicable form of compressor which achieves perfect isothermal compression. There are at the present time several large installations of this type that have been operating in a satisfactory manner for years. They are all built on the so-called Taylor system which gives high efficiencies and of which the operating expenses are quite moderate. As a permanent installation where high efficiency and low operating cost can offset a somewhat high initial cost the Taylor compressor is eminently satisfactory, but it is not suitable where a compressed-air plant is required for a relatively short space of time as, for example, in construction work.

The present investigation describes an apparatus working on the jet or injector principle, which is supposed to supply the field where the Taylor system is not applicable. The earliest known form of hydro compressor was the trompe or water bellows, which was used by the Romans to provide the blast required in the operation of forges. In 1861 Sommeiller constructed a number of compressors operating on the principle of the ram and applied them with great success in the construction work of the Mont Cénis tunnel. Later on H. D. Pearsall and J. P. Frizell experimented with various types of compressors. The latter was the first to realize clearly that the pressure developed is dependent on the height to which the water is carried up after passing through the separating chamber.

The author's work was confined largely to the jet or injector compressor. The basic principle of this compressor depends on the mutual convertibility of pressure and kinetic energy in a closed conduit under conditions of steady flow, and the faculty possessed by a fluid of entraining air when moving at a high velocity. As pressure and velocity are functions of the cross-sectional area of the conduit, any desired conversion, within limits, can be made by appropriate variations in the cross-section. Air can be introduced at a region of low pressure, and discharged at one of high pressure.

The conversion of pressure into velocity can be effected by reducing the cross-section of the conduit, either gradually in a nozzle, or abruptly by means of an orifice. The issuing jet, being in communication with the atmosphere, entrains the air immediately surrounding it, and by reason of the turbulence existing in the jet a mixing or emulsion of air and water ensues. The combined jet then flows into a diverging channel where its kinetic energy is gradually transformed into pressure. The air may be recovered from the water under pressure by allowing the mixture to flow into a separating chamber of large cross-section, where the velocity will be sufficiently reduced to allow the bubbles of air to float up and disengage from the water. Air can then be drawn off from the top of the chamber, and the water allowed to escape through a valve located at or near the bottom.

An analysis of the transformations of energy given in detail in the complete thesis brings out several interesting facts, one of which is that the efficiency of a nozzle is equal to the square of the coefficient of velocity and has no connection with the coefficient of contraction. The loss to abrupt expansion is also given by a formula in the unabridged thesis.

A number of experiments have been performed with diverging cones, using various angles of divergence; and it has been found that the loss of energy is practically constant for all angles of divergence from 2 deg. up to 5 deg. 30 min., but increases rapidly above this limiting value. Experiments made by Dr. Stanton on a cone having an angle of divergence of 5 deg. showed a loss equal to $0.2\ v^2/2$ g., where v is the velocity at the throat.

These results, while useful in themselves, are not applicable to the design of diverging channels with other than straight sides. It appears highly probable that the cone is by no means the best form of diverging channel.

[1] Abstract of a thesis submitted in June, 1925, to the Department of Mechanical Engineering and to the Committee of Graduate Study of Stanford University in partial fulfilment of the requirements for the degree of M.E.

The tests described above have brought out the fact that the jet compressor can be used to obtain relatively high compression pressures, the best results being obtained with a compression pressure equal to 50 per cent of the supply pressure; but the efficiency is very low, not exceeding 15 per cent for a single stage. With such a low efficiency it is questionable whether compressors of this type could be used advantageously except under exceptional circumstances, namely, when there is an abundant supply of water under a head of at least 450 ft. With such a supply head air could be compressed to a pressure of 100 or 120 lb. per sq. in., which is a sufficiently high pressure for most pneumatic tools.

If a piston compressor driven by a gasoline engine were used the combined efficiency would not be likely to exceed 11 per cent, assuming a thermal efficiency for the gas engine of 16 per cent, and an isothermal efficiency for the compressor of 70 per cent. The jet compressor with an efficiency of 15 per cent compares favorably with this combination, particularly in view of the fact that water in large quantities can often be had for nothing, or at very small cost.

Another alternative is to use a turbine or impulse wheel directly connected to an air compressor. In such a case the combined efficiency would be in the neighborhood of 60 per cent, but the first cost would be very much higher than for a jet compressor, and the question of choice resolves itself into a problem in economics.

Discussion of Results

Test No. 1. The graph of this test shows that there is a linear relation between air capacity and compression pressure, the volume of air varying inversely with the compression pressure.

The variation of capacity with supply pressure for three values of compression pressure is shown by curves, from an examination of which it appears that the capacity varies approximately with the square root of the supply pressure; this means that the capacity, for a given compression pressure, varies directly with the velocity of the jet; for $v = \sqrt{2gH}$ and the head H is a direct function of the supply pressure.

Tests Nos. 2 and 3. These tests, performed with the "complete nozzle assembly," showed a distinct improvement in air capacity as compared to the "single nozzle assembly," the difference being more marked at the higher compression pressures.

Curves given of the characteristics of the complete nozzle assembly show very low efficiencies. This is primarily due to the large hydraulic losses resulting from the abrupt expansion and the flow through the diverging channel; another factor of great importance in reducing the efficiency is the falling off in capacity with increasing compression pressure.

Tests Nos. 4 and 5. The results of these tests made with the "flaring jet orifice" showed no improvement over results obtained with a plain jet. Apparently the lower hydraulic efficiency more than offset the added capacity. The maximum net efficiency obtained with a supply pressure of 50 lb. per sq. in. was 17 per cent, and under the same conditions the "complete nozzle assembly" gave an efficiency of 23 per cent.

Specialization is the moving spirit of the age.... No single small unit operated as a mere incident to a railroad or industrial establishment can make the fullest improvement in quality nor aid materially in perfecting the art or in decreasing the cost of manufacture.

The remarkable industrial development in America as contrasted with Europe is due largely to the fact that the American people have been economically able to manufacture and purchase in large volumes. Larger business units necessarily mean easier absorption of overhead expense, and allow leeway for expenditure of time and money in improving the product, cheapening cost of production, and lowering the price to the consumer. (The *Iron Age*, August 5, 1926, p. 360.)

Group Drive and Individually Motorized Drive

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THE author consented to prepare the present paper with great reluctance because of the number of viewpoints from which this particular subject may be considered. Much depends upon operating conditions existing in the plant under consideration. One individual will place considerable stress upon the elimination of certain details; the next will feel that these particular points are not so important as certain others, and hence, within the space limits available, it is difficult adequately to cover the subject.

The experience of the author in the motor-application field inclines him to the belief that, unless all of the operating conditions are known, it is difficult to decide which would be the better of the two methods, excepting for certain specific applications. There can be no doubt that the individually motorized tool is gaining in favor for a great many, if not most, applications. There can be little doubt, however, that for a miscellaneous collection of tools, such as frequently exists in medium-sized machine shops, both types of drive have their place, but what might be particularly suitable in one shop might not work out nearly so well for another plant where the operating conditions were somewhat different. Under some circumstances, therefore, it seems that the use of an individual motor for each tool is the proper application, while with other conditions, the group drive represents at least fairly good practice. There are a great many factors entering into the problem, and the author feels that only when all of the conditions are known can the correct decision be made.

In the following, the author will attempt to present a number of the advantages and disadvantages of the group as well as the individual-drive system.

Individual Drive for Shops Machining Heavy Work

The shop that has very heavy or large work, such as is encountered in the machining of locomotive parts, turbine-condenser bases, or large alternating-current generator fields, should be equipped with individually motorized units. In such shops the overhead crane of large capacity is an absolute necessity, and overhead shafting is therefore out of the question. Overhead belts or shafts in practically any form interfere with the crane service because the pieces are so large they cannot be hoisted high enough to clear, even if the space between units permits the raising and lowering of the work. Another factor for such shops is the fact that the main-drive units are usually so large that it is not particularly feasible to group-drive the tools in any case.

The portable tools used to machine large, heavy castings are of necessity individually motor-driven. In these cases the tool is moved to the work and a number of set-ups and changes in position of a large heavy casting are avoided. One set-up on the iron floor suffices.

It may be said that equipment demanding a heavy continuous load, i.e., 25 hp. or more—can usually be individually driven to advantage. Many smaller loads place themselves in this category as well, such as small pumps, some fans, or cases where there are unusual conditions of speed, as is frequently the case with centrifugal compressors.

Woodworking Machinery

Present-day practice in the woodworking establishments seems to preclude the group drive for practically all of the production tools. The power required per spindle and the speeds necessary to meet competition today mean not only individually motorized tools but individual motors for each spindle. Large planers and matchers have as many as eight motors, and not infrequently the top head will have two motors, one on each end, and each may be 25 or even 35 hp. All motors, excepting the feed motor, will operate at 3600 r.p.m. It will be obvious that such conditions form a decided barrier to belt drive when powers of this order are required at such high speeds. The molder is also another good example in this class of tools. Here we have four head motors operating at a minimum of 3600 r.p.m., and frequently up to 6000 r.p.m. The rating of motors so applied will vary from 5 to 25 hp. at the low speed. Another very good example is the modern shaper, which normally operates at 7200 r.p.m. and is very frequently equipped with $7\frac{1}{2}$-hp. motors on each spindle. A brief consideration will show the exceedingly high belt speeds which would be attained if attempts were made to belt such equipments, to say nothing of the fact that a very considerable addition in floor space would be required. In densely populated locations floor space is certainly becoming an increasingly important factor, due to the high rental charge therefor. In addition, motors of the class ordinarily used in the woodworking industry are normally designed to have a very high maximum or momentary breakdown capacity, and this over-capacity is frequently taxed. Hence the necessity for individual motors is very evident.

Grinding Machinery

The grinding field is daily becoming of greater importance in the individually motorized field. The ball-bearing industry has developed to a tremendous extent. The grinding of the races is a very important part in the manufacture of such equipment. Small curved surfaces must be accurately ground to size. Grinding wheels of small diameter are a necessity to secure the small radii which are required by the ball-bearing units. Small diameters in turn mean high rotating speeds if the proper surface speed between work and wheel is maintained. Belts have been used to a considerable extent in this field, but even with the small amount of power required by the small wheels, when belt speeds exceeding 6000 ft. per min. are reached, unsatisfactory operation is very apt to creep in. A small high-speed alternating-current motor of the squirrel-cage type mounted directly on the spindle forms a very satisfactory solution for this particular problem. Speed variations of any particular importance, such as would be met with by belt slippage, are practically eliminated. For the larger grinding jobs, again an advantage is obtained. Instead of having a very wide and cumbersome structure in which a belt slides back and forth, it is a very easy matter to arrange a flexible cable to the unit so that the motor can travel with its wheel and be traversed the whole length of the machine without a cumbersome overhead structure. Installations involving motor speeds up to and including 25,000 r.p.m. are in active operation today. A moment's calculation will show what this would mean if an attempt were made to drive by belt. A pulley 2 in. in diameter gives a belt speed of more than 13,000 ft. per min. In such a case maintenance would be very high and operating difficulties would be considerable.

In the foregoing, consideration has mainly been given to strictly modern tools engaged on high-speed production jobs. Exceptions may be found in nearly all, if not all, industries, especially those under the class of job shops. The larger drives, such as are frequently employed in the steel-mill industry, preclude the possibility of using anything except a chain or a direct drive, the horsepowers involved frequently being far beyond any belt capacity. Such drives, of course, come under more or less special classifications along the lines outlined for other special industries.

Difficulty of Proper Ventilation and Light, and of Shifting Tools in Group-Drive Systems

A decided disadvantage in connection with group drive and one which is receiving more and more consideration is the fact that better ventilation and better lighting conditions are possible when individual drives with the motor mounted on the driven machine are employed. In the automotive field, where 24 hours of work per day is so frequently the rule, this is of great importance. The lighting conditions must be of the very best if anything like efficient production is to be secured during the hours of darkness, and even during daylight hours on cloudy or dark days a considerable amount of artificial lighting is usually necessary. A maze of belts practically

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precludes the possibility of securing shadowless lighting. It is recognized that without adequate lighting, neither day- nor night-work production will be up to standard. Another factor which is also creeping in is the attention which the various safety organizations and state laws are giving to the subject of having the worker adequately protected and working under healthful conditions.

In one prominent plant in the automotive field the planning section had caused a rearrangement of groups of production tools upon several occasions. Group drive for the major portion of the machines prevailed and a very large majority of the tools were of the belt-driven type. A heavy expense would have been involved in the shifting of lines of shafting and the realigning of tools. It was very apparent that the purchase of complete new tool equipment would be prohibitive, especially as a considerable portion of the tools were in first-class shape. In the face of such conditions and the tremendous outlay for new equipment, a compromise was decided upon. The oldest and least productive tools were replaced with new equipment, which were individually motorized. The remainder of the equipment was graded with the idea of replacing it from time to time as the occasion arose. To take care of the immediate necessities, a considerable number of the tools were modified to provide for individual-motor drive, this in spite of the fact that it usually does not pay to spend much to modify a tool and change it over from belt to individual motor drive. A further complication existed at this plant because of the fact that considerable floor space was available which would not, due to the light sidewall and overhead structure, support lines of shafting. Obviously, the motorized tool which does not require any overhead structure would answer this particular part of the problem. The author happens to know that this program was carried out and that at the present time a very large proportion of the production tools have been replaced by individually motorized units in the place of the older belt-driven tools.

There can be little or no discussion upon the point as to the expense of applying individual motors to each tool, nor can there be any doubt but what the connected horsepower load of the motors involved will exceed that required for group drive.

Influence of First Cost

To how great an extent the first cost will be increased depends very largely upon the class of the tools and their size, as well as the number of motors involved to drive properly the operating heads or spindles. The answer to the question of first cost, it seems, must come either from the location in which the tool is to be placed or else upon how much more production can be expected from the individually motorized unit. The management of any industrial plant is going to be rather cold-blooded when it considers from a dollars-and-cents point of view an increased first cost unless it can be clearly shown that there is going to be a very decided advantage either directly or indirectly. If the tool is going to be located in a place where crane service is imperative, obviously the choice must be in favor of the individual tool, or else more will be lost than gained in the servicing of the tool. In the larger industrial plants it not infrequently occurs that certain jobs must be put through on a rush basis, which in turn means overtime or night work and, unless there is a night gang working, the day force must stay over to complete the emergency job. The individually motorized tool here obviously has its advantage.

Some of the arguments presented in favor of the individual tool are the elimination of belt and lineshaft losses. This is a very frequently presented argument for the individually motorized drive. The stressing of this point can readily be carried too far as the savings are more apt to be of a decidedly minor nature. Lineshafting makes a considerable show because it is conspicuous, but except when new and stiff, as when perhaps new bearings or something of the sort have been installed, the actual power required to drive a reasonably well-lined-up lineshaft is small. Moreover, against the larger motor driving a group, we have a number of the smaller individual units which, by themselves, are not so efficient as the one larger motor. Therefore, if due care in the group selection has been made, the reduction in the operating cost due to power saving as compared to individual drive will be relatively small.

In this connection it would be well to bear in mind how very small the cost of power per unit is in the average industry—one-half to two per cent being a common figure. Even if it were entirely eliminated the final cost per piece would not be appreciable. It seems, therefore, that while no manager would belittle a saving, this is not a particularly vulnerable point of attack.

A much better point of attack from the group drive would be along the lines of the greater installed motor capacity which is required of the individually motorized unit as compared to that required for the group drive. From 5 to 7 to 1 is probably a reasonably good increase in horsepower—i.e., connected load. In short, by careful selection it is possible, at least in many plants, to reduce the actual connected horsepower of the motors involved from one-seventh to one-fifth of that required if individually motorized tools are required.

Many vs. a Few Motors

It is well known, moreover, that a number of motors, aggregating a given total, cost far more than does one large unit having the same capacity. The same is also true of the control. Maintenance of motor and control is also reduced. For many industrial applications it may be said that the larger the motor, the less likely it is to give trouble provided it is reasonably well-looked after. The answer to this, of course, is the fact that advantage can be taken of the diversity factor and possibly of the production routine when group-driven units are employed. When an individually motorized tool is used, each and every tool must be motored to its full capacity and to meet maximum operating conditions. By a careful selection of tools, a very large diversity factor can be taken advantage of, and it is not likely that more than two or three tools in the group would be making maximum demand at a time. There is always a considerable time required for the finishing cut, which is usually light, and the setting up of work and its removal from a machine, all of which in a group drive can be taken advantage of to help the diversity factor; and even if the group motor were temporarily overloaded, the conditions would not likely exist long enough to work any injury to the motor equipment. Hence an advantage in first cost can be secured by a careful selection of the tools and of the routine work flowing through the shop. The manufacturers of many tools lay great stress on the powerful motors with which they equip their machines, but there is little virtue in this of itself, especially in factories where alternating current is employed, because the tools frequently work for long periods at fractional loads and possibly never get to their actual maximum operating capacity, with the result that there is a relatively large motor employed working at fractional loads all of the time, resulting in low efficiency and low power factor. On an individual motor possibly this is of minor importance, but when one considers a shop that employs several hundred motors, each of which is performing more or less in a similar manner, the plant engineer must be alert or he will penalize his company, because a large load of low power factor will mean an increased price for the power purchased; or, on the other hand, if the power is generated at the plant, more generating equipment for a given actual kilowatt load will be required. Either way it is figured, the same result will be obtained—either a higher price must be paid for power or else more generating equipment must be provided, and this immediately reacts against any possible savings obtained at the power house.

It is recognized that it is by no means an easy task properly to arrange a group of tools and at the same time secure the maximum operating advantages. A great deal of skill and planning experience should enter into such an arrangement, and it may mean that, after a careful tryout, a rearrangement of the groups could very profitably be made.

Due to the fact that production requirements are continually changing in most shops, it may also prove that the shifting of production should also shift the grouping of machines, at least as regards the amount of power required by certain units. Obviously, if there were very much shifting of this sort, the actual cost to shift the machines from one group to the other in order to drive them economically would have to be considered, and this might prove to be disadvantageous.

Diversity of Motors

A factor which may have a very important bearing on many industrial plants is the great diversity in the number of motors

involved when the individually driven tool is employed. For example, in the woodworking industry, due to the fact that very close centers must frequently be provided for, a special type of motor has been developed by a number of the leading motor manufacturers. While such machines give a very high production and an excellent quality of finished product, as far as the motor equipment is concerned, variety of special motors is decidedly objectionable.

This is true to a certain extent as regards machine tools used in the metal-working trade, but not so much as in the woodworking industry. The common practice of machine-tool builders of supplying such motors as their selling agents may determine and which are not specifically demanded by the user otherwise, leads to a very great diversification in the motor equipment that is likely to exist in a given plant. Obviously, this presents a very decided disadvantage in that a very large number of spare parts of various sorts must be kept on hand if a reasonable safety factor is employed. The alternative is to provide a number of spare motors, but for a large proportion of the drives it is rather prohibitive in cost to provide very many spare complete units. The same argument also holds in the case of the control.

One prominent manufacturer of a motor equipment has recently reduced the number of spare bearing linings which will be required if his motors are used. In fact, such motors as are considered as being in the industrial class up to about 200 hp. require only thirteen bearing linings in both alternating- and direct-current types. Obviously this will greatly simplify and reduce the stock or spare-part problem to a marked extent. This is also insurance against the chance of not having the particular bearing lining needed, because an adequate stock of spares can readily be maintained without having a large sum tied up in inventory.

Cost of Wiring and Control Apparatus

In considering the first cost of installation of the group versus the individual motor drive, there is another very great advantage and that is in the wiring. The elimination of a number of switches, outlet boxes, branch circuits, etc., to say nothing of the control units, permits a very great reduction in the installation cost, to say nothing of the time that it takes to install such equipment. Of course, it must not be forgotten that over and against this is the cost of installing the overhead lineshafting, jackshafts, and possible intermediate lineshafts which may be required. How much this will offset the cost of the wiring or exceed it depends on the distances and the difficulty in lining up the lineshaft, as well as on its first cost.

Synchronous-Motor Drive

In addition to the possible advantage which will accrue in the group drive, synchronous motors may be employed to drive various groups, and this will greatly aidt he power-factor situation where alternating current is employed. Synchronous-motor costs have been greatly reduced recently, which materially assists in helping to solve this difficult problem. The maintenance of a reasonable power factor will usually secure a lower power cost, in addition to making it possible to put in smaller copper in the building, and will materially decrease the line regulation for the copper installed. It matters little whether or not power is purchased or generated by a given plant. The net or final result is practically the same. If the power factor be maintained at a reasonable value—say, 80 to 90 per cent—the actual cost to the user thereof will be less in a very great majority of the cases. This question is becoming increasingly more important all the time. Most power contracts today contain a penalty clause (possibly not at present enforced) for low power factor, and frequently such contracts also contain a clause which gives a bonus for power factors maintained above certain values. These vary in different localities, but as stated in the foregoing, the advantages which would accrue through having an 80 per cent power factor or better are, in effect, the same whether the power is purchased or generated.

It is true that practically all of the major motor manufacturers have improved their products to a marked extent in the last few years. Nevertheless it is possible that the routine maintenance and inspection of electrical equipment is greatly reduced through the use of group drive, and it is likely that using larger units—say, 25- to 50-hp. motors—will result in less maintenance electrically than could be expected were a larger number of small motors employed.

Servicing of Electrical Equipment

It is appreciated that the actual servicing of equipments varies greatly in individual plants. It has been the author's observation that, while many manufacturing plants will carefully supervise the inspection of their machines and possibly the motors, yet the control is all too frequently neglected. A plant which will very carefully go over its motors to see if the bearings are oiled and possibly the commutator is in good shape and that the brushes are in good order, will overlook the control and give it no attention whatsoever. This neglect in the past has resulted in much unnecessary delay. Just why the users of control equipment seem to feel that it will get along without any maintenance or attention is something which electrical manufacturers to date have not been able to discover.

In a group drive a careful selection will greatly reduce the number of different sizes of motors which are required for a given plant. It is common practice for certain individual plants to standardize on certain reasonably sized units and to make use of such units, although sometimes they will be underloaded, at least for certain periods. The very great virtue of this plant is that a few spares only are carried, and another unit can be put in place and no great amount of time or loss of production is entailed.

The situation as to the variety of motors could, of course, be greatly improved if manufacturers were to standardize upon some one particular make and insist that all of the tools coming into their plant be equipped with motors of a certain manufacturer.

Data on Maintenance Cost Misleading

It seems to the author that many of the articles which present data in connection with maintenance cost of motors and control show a relatively high maintenance figure, especially when consideration is given to burned-out motors. Under most operating conditions where maintenance figures are given it would seem that either there is something fundamentally wrong with the application or else with the motor which has been used. It is true that motors frequently will be abused in meeting operating conditions temporarily, either due to emergency or some other unforeseen cause, but in general, if motors were applied properly as to their horsepower and their general characteristics, the maintenance cost could be greatly reduced. This particular phase of the application has been brought out in a number of instances in which motors which were not suited to the application have been used or else have been greatly overloaded for one reason or another. The expense so incurred is obviously not properly chargeable against the motor. A point has been made of the insurance which a drive provides—i.e., against an undue number of shutdowns. In a highly organized plant a short delay means a considerable loss in production, possibly extending even to the shipping department. However, with skilful and trained help provided with adequate repair facilities, it seems as if there should be little difficulty in avoiding shutdowns of a duration long enough to be detrimental to the total production for the day. Steel mills are probably better organized than most other industrial plants, although at the present time the more progressive companies in the automotive industries are actively engaged in placing their maintenance departments in the very best of condition. On the other hand, there can be but little doubt that there is less chance of being able to standardize on a few sizes of motors for individual drive than there is in case group drive is employed.

Selection of a Proper Motor for a Given Application

The author has made a point of selecting the proper motor for a given application. It is true that in a number of instances motors have been selected by plant engineers and standard units specified in an effort to standardize on the number of units required. In some instances undoubtedly this has led to a higher maintenance cost than otherwise would have been necessary. In this connection also, it would be well to refer perhaps to the recommendations called for by some machine-tool builders, in which motors have been specified which are really much larger than seem to be required for the job, particularly if proper consideration is given to a motor having the correct characteristics for the work it is called upon to do.

Punch presses and shears are notable examples in this connection.

All too frequently the standard general-purpose squirrel-cage motor has been employed, even though motors having the proper characteristics are available which would perform the work much easier. Motors of smaller horsepower could usually be employed if those having the proper characteristics were used. One electrical manufacturer recently had an order from an automotive manufacturer who was adding a large number of tools in his program of plant expansion. The recommendations from the manufacturer of the tools as to the motors were also secured. In going over these recommendations it was the belief of the motor manufacturer's engineers that in a great number of cases the capacity of the motors could be reduced to 60 or 75 per cent of the recommended horsepower capacity. It was only after a considerable delay had ensued and careful study been made of the situation that the ultimate user consented to accept the recommendations of the motor manufacturer. It is now a matter of record that in only one instance in which the recommendations of the motor manufacturer were followed has any trouble ensued, and it is not certain even in this one instance that the motor selected was too small for the job. The saving in first cost to the ultimate user, of course, was a relatively large one, and, in addition, he is now in a position where his motors are reasonably well loaded up; and a report has been recently received to the effect that the power factor in his system averages around 80 per cent, which, in an industrial plant, is good. Another factor which the author believes to be of considerable importance in connection with group versus individual drive is the fact that in industrial plants a considerable percentage of the personnel employed are mechanically trained, while on the other hand only a very few of them are in any way familiar with the electrical end of the drive of their particular machine. The result is that defects possibly due to belts, lack of oil, etc., will be very noticeable and will be taken care of by the individual operating the machine; on the other hand, the electrical difficulties of the motor are allowed to go until trouble develops, which if taken care of at the time it first occurred would have been of a very minor nature, yet when allowed to develop, has ultimately cau ed the motor to be replaced, with more attendent expense and delay than would have been necessary had the trouble been eliminated in the first place. It is a common remark to hear a man say, "Well, I know nothing about electricity so I didn't do anything about it," with the result that in the end he loses by not calling the attention of the electrician to the difficulty, hoping that the trouble will clear itself. Such a condition may be described as being somewhat like that of an infection; if taken care of promptly the results are minor or negligible in their effect. However, if the infection is allowed to spread, there is but one result, that finally the person infected must consult a doctor or go to a hospital and be out of work for perhaps a considerable period.

A considerable point has sometimes been made of the possibilities involved in being able to motorize a group right up to its capacity and utilize the motor at just about its normal load, or perhaps take advantage of its overload capacity temporarily. It does not seem to the author that this presents any particular advantage to either the group or the individual drive, and that the advantages to be gained in the one case are just as great in the other. As he sees it, one very great disadvantage to the group drive is the fact that in the lining up of a group of machines to be driven by one motor, it automatically includes the thought that each of the machines is going to be running most of the time, although probably not at capacity. It is upon this assumption that most of the power-consumption figures offered in an attempt to show savings have been made. It is well known, however, that in many shops where group drive has been installed possibly only one machine or two out of a group of six or eight or more is employed. The author had occasion to make a survey in two plants of one rather extensive industrial concern in which a very large number of group drives were installed. In spite of the fact that lineshafting prevailed to a marked extent, it was found that in one of the plants, which had a connected load of 6600 hp., the actual power demand was about 10 per cent of the connected load, and in the other, where the connected load was approximately 7600 hp., the actual power demand was only about 9 per cent, which would indicate that by no means can we expect to have the various motors completely loaded up to their capacity, even though the most approved type of group drive is employed. Another advantage for the individual drive, especially in large machines, is the ability to start and stop from any number of points, inasmuch as magnetic control can be employed and as many push buttons or controllers employed as may be required. In case there is a shipper rod, this is not always a feasible procedure.

Ease of Control of Individual Motor Drive

Magnetic control tends to reduce the abuse to which motors are more or less subjected when starting and stopping frequently. The magnetic control automatically causes acceleration always at the same rate, no matter how ignorant the operator may be or whether he uses a broom handle with which to start and stop. The result to the equipment is always exactly the same.

Speed controls in connection with such machines as require close speed adjustment with the individual tool are decidedly in an advantageous position. The modern adjustable-speed direct-current motor can be run through a wide speed range, with increments or decrements which are scarcely perceptible if such fine adjustment is necessary.

Summary

In summarizing, it may be said that group driving will not go out of use unless there are very unexpected developments in the motor field. It is not felt that there is now or likely to be any hard-and-fast rules which will prevent the installation of either the individual or the group drive. There are too many factors involved, all of which must be given the proper weight when it comes to making a decision of this character. There are advantages and disadvantages to either of the two systems, and the former must be weighed against the latter as to what will be the result in the way of final cost at the end of the year. There are of course certain industries in which it would seem perhaps that group drives are unusually successful. The textile industry is perhaps a good example of this. However, in some other industries, notably the steel and machine-tool, the tendency is most certainly toward the individual-type drive. Generally, the author thinks that at the present time individual drives seem to predominate—i.e., as far as newly installed equipment is concerned. However, as pointed out earlier, it does not seem as if in most cases the actual friction losses which would be eliminated through the use of individual drive in themselves would carry enough weight positively to swing the decision toward either one or the other of the two types of drive.

It has also been pointed out that in many of the states the safety laws are decidedly against the installation of large groups in which lineshafts and jacks are carried overhead. The vertically running belts are a continual hazard not only to safety in operation, but also in the cutting off of light and ventilation in the room; and in order to have the very best output from an operator, it is necessary that he be working under good operating conditions.

One factor which has not been particularly stressed in the foregoing is that, if the plant management is sufficiently interested to have some one appointed to go throughly into the problem, the various operators can be trained to operate their respective machines with a view to getting the most out of them with the least power cost. If first cost is of extreme importance, group drive offers a much cheaper solution than the individual drive.

Not always can the management choose the particular building in which it must produce a given article. Frequently the floor space in a building will be quite adequate, but the overhead construction will not admit of lineshafting. In such a case, obviously, the individually motorized unit must come into its own.

Another factor which has not been mentioned in this connection is that in arranging tools for group drive it is frequently very much more convenient, as far as the product is concerned, to place the tools one way, while a much better tool arrangement, as far as tools are concerned, could be secured another way. In such cases the ideal arrangement, as far as the motor drive is concerned, must be sacrificed in order to meet the proper flow of production, because it is not a feasible thing to have production jump forward and back and around, but rather should it flow in a continuous stream. The material should enter at one point and progress to where the finished product emerges. Still another point that has not been stressed is the fact that in locating groups with overhead lineshafting and vertical belt drive, the machines must be carefully lined up

(*Continued on page 969*)

The Influence of Elasticity and Errors in Tooth Shape on Stresses in Gears[1]

Progress Report of the A.S.M.E. Special Research Committee on Gears

By JOHN EDWARD NICHOLAS,[2] CAMBRIDGE, MASS.

RECOGNIZING the timely importance of a study of the effects of varying degrees of tooth accuracy and varying velocities on the strength of gear teeth, the Main Research Committee of The American Society of Mechanical Engineers with the approval of Council organized a special research committee to make such a study. This committee held its first meeting in April, 1922, and requested Wilfred Lewis to serve as chairman and to begin the systematic development of the design for a second gear-testing machine for which he had already made rough sketches. From time to time the members of the committee recommended modifications of the original drawings.

The machine was built, and was exhibited at the Annual Meeting of The American Society of Mechanical Engineers in December, 1924. It was then shipped to the Massachusetts Institute of Technology where the actual tests were to be made under the direction of Prof. E. F. Miller of the Department of Mechanical Engineering. Upon its arrival, the machine was set up and certain changes in it were found advisable. The present paper is abstracted from a thesis which describes the first series of tests and the results obtained. For those unfamiliar with the machine, the following brief description is inserted.

Description of Machine[3]

The purpose of the Lewis gear-testing machine is primarily to determine the effect of varying degrees of tooth accuracy and varying velocities on the strength of gear teeth.

The machine consists of a pair of test gears and pinions mounted in a swing frame centered in the axis of the pinion shaft and supported at a convenient distance upon weighing scales. By this means the torque can be measured when transmitting power received from a belt on a flywheel pulley mounted on the pinion shaft. The test gears are mounted on a telescopic sleeve and shaft, in which an initial torque can be introduced through a connecting nut of long pitch at the outer end of the telescope. The elastic reaction of the telescope maintains any desired average load on the gear teeth, which is augmented and reduced in action by the inaccuracies in the teeth and by the speed. Insulated ball bearings are used throughout, and electric circuits are established through telephone receivers which may be interrupted when either test gear fails to make contact with its pinion. When starting, and at slow speeds, there will be no interruption, but as the speed is increased there will come a time when the inaccuracies cause a momentary break in tooth contact. This will be announced by one of the telephone receivers and at that moment the corresponding speed, as indicated by the tachometer, is noted. Another initial load will show a different breaking speed, and in this way the loads and speeds can be correlated by experiment. For rigid teeth in rolling contact the initial load required to maintain contact is proportional to the speed squared, and the departure from this relation may be ascribed to elasticity and mass effects. For the accurate determination of the latter, special provision has been made in the application of a torsion balance to the pinion shaft direct. A light rod of tempered steel is attached to the pinion shaft and anchored at its outer end. A capillary pen mounted in a holder traversed by the oscillation of the pinion shaft moves over a paper ribbon traveling at a known rate of speed. This device was designed by H. H. Williams. It makes possible the accurate determination of the oscillating period for the pinion shaft alone, with and without one or two of the flywheels, and with or without each of the test gears.

Provision has also been made to multiply and record the inaccuracies in the teeth on circular diagrams in which they will appear as radial displacements, and a novel feature of this mechanism is that diagrams can be made and compared for the same teeth under very heavy loads as well as under the ordinary comparatively light loads which cannot show the effects of elastic deflection and compression.

The Charts

Reproductions of the charts showing the combination of gears used in the tests reported in this paper may be found in Figs. 1 to 4.

B to H, Fig. 1, are charts of steel pinions meshing with steel gears.

I to K, Fig. 2, are charts of steel pinions meshing with cast-iron gears.

L and M, Fig. 3, are charts of cast-iron pinions meshing with cast-iron gears.

The arrows on these prints show the direction of rotation of the gears, whereas the tooth marked No. 1 is the reference tooth from which all other teeth are located. The tooth marked with an asterisk is the tooth of maximum error. By measuring the maximum radial distance along the tooth of maximum error in inches and dividing by the factor 250, the maximum error for that combination is found. The constant 250 is the lever arm of the multiplying linkage in relation to movement at the pitch line. Outward movement indicates acceleration of the driven gear, while inward movement indicates retardation of the driven gear.

Chart N, Fig. 4, represents the composite errors of the master set alone which were used in the comparison of all other combinations.

TABLE 1 GEARS USED IN THE TESTS

Symbol	Diametral pitch and number of teeth	Material	Normal or specified error, in.	Lewis y factor
GX3-18-1	3DP-18T	H & G steel	±0.0003	0.120
GX3-18-2	3DP-18T	H & G steel	±0.0003	0.120
GX3-18-3	3DP-18T	H & G steel	Spacing ±0.0005	0.120
GX3-18-4	3DP-18T	H & G steel	Spacing ±0.0020	0.120
GX3-18-5	3DP-18T	H & G steel	Normal pitch −0.0005	0.120
GX3-18-6	3DP-18T	H & G steel	Normal pitch +0.0005	0.120
GX3-18-7	3DP-18T	H & G steel	Normal pitch −0.0020	0.120
GX3-18-8	3DP-18T	H & G steel	Normal pitch +0.0020	0.120
GX3-18-9	3DP-18T	Cast-iron, generated	±0.0010	0.120
GX3-18-10	3DP-18T	Cast-iron, form milled	±0.0040	0.083
GX3-48-1	3DP-48T	H & G steel	±0.0003	0.126
GX3-48-2	3DP-48T	H & G steel	±0.0003	0.126
GX3-48-3	3DP-48T	Cast-iron, generated	±0.0010	0.126
GX3-48-4	3DP-48T	Cast-iron, form milled	±0.0040	0.112

TABLE 2 ERRORS AS DETERMINED BY INDICATOR, ODONTOMETER AND CHART

Run No.	Pinion	Gear	Max. error with ind.	Odontometer error	Normal or specified combined error	Chart error	Assumed error
A	GX3-18-8	GX3-48-2	0.0027	−0.0020	0.0023	0.0007	0.0025
B	GX3-18-7	GX3-48-2	0.0021	+0.0020	0.0023	0.0015	0.0015
C[1]	GX3-18-8	GX3-48-2	0.0027	−0.0020	0.0023	0.0007	0.0025
D	GX3-18-4	GX3-48-2	0.0043	±0.0020	0.0023	0.0021	0.0021
E	GX3-18-2	GX3-48-2	0.0021	±0.0003	0.0005	0.0005	0.0005
F	GX3-18-3	GX3-48-2	0.0024	±0.0005	0.0008	0.0006	0.0008
G	GX3-18-6	GX3-48-2	0.0021	+0.0010	0.0013	0.0005	0.0012
H	GX3-18-5	GX3-48-2	0.0017	−0.0007	0.0008	0.0008	0.0008
I	GX3-18-2	GX3-48-3	0.0029	0.0006	0.0013	0.0021	0.0021
J	GX3-18-6	GX3-48-3	0.0024	0.0010	0.0020	0.0021	0.0021
K	GX3-18-7	GX3-48-3	0.0029	0.0020	0.0030	0.0026	0.0026
L	GX3-18-9	GX3-48-3	0.0033	0.0012	0.0020	0.0023	0.0023
M	GX3-18-10	GX3-48-4	....	.....	0.0040	0.0033	0.0033

[1] Same as run A

[1] Part of a thesis submitted to the Massachusetts Institute of Technology, Cambridge, Mass., for the degree of Master of Science. The work reported was done under the direction of Prof. C. E. Fuller and Prof. Earle Buckingham, member of the Special A.S.M.E. Research Committee on the Strength of Gear Teeth, and has been arranged for presentation by Professor Buckingham as a progress report of the Committee at a Technical Session, Friday morning, September 10, 1926, in connection with the Sixth Annual Machine Tool Exhibition, Mason Laboratory, Yale University, New Haven, Conn.

[2] Assistant Instructor, Massachusetts Institute of Technology.

[3] For details see *American Machinist*, Dec. 13, 1923.

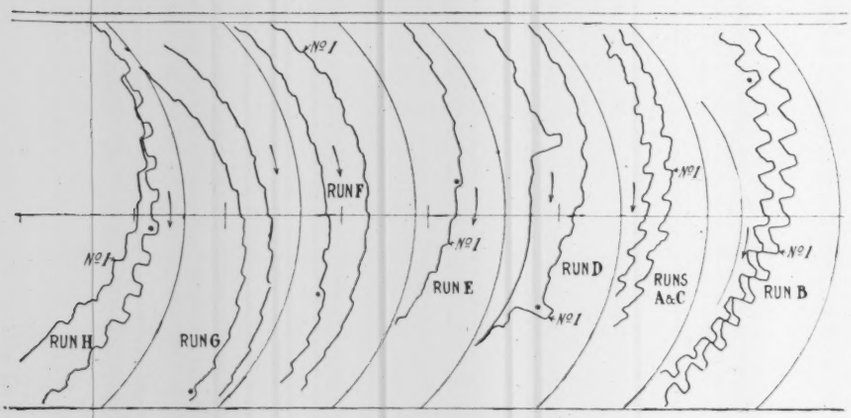

Fig. 1 Charts of Steel Pinions with Steel Gears, Runs B to H, Inclusive
* Indicates tooth of maximum error.

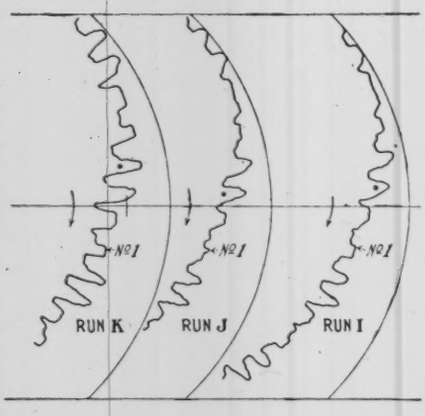

Fig. 2 Charts of Steel Pinions with Cast-Iron Gears, Runs I, J, and K
* Indicates tooth of maximum error.

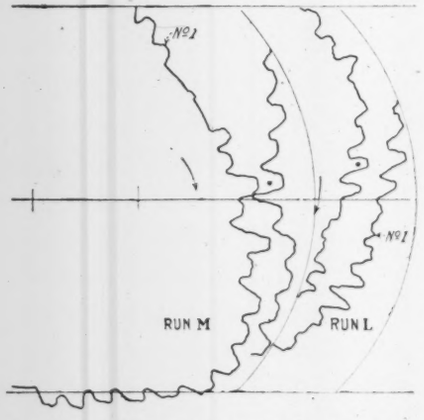

Fig. 3 Charts of Cast-Iron Pinions with Cast-Iron Gears, Runs L and M
* Indicates tooth of maximum error.

It shows four complete revolutions of the gear.

Table 1 is a tabulation of the gears used in the tests.

Table 2 is a tabulation of combinations used and the errors of each set.

Determination of Errors

An attempt was made to check up the chart errors both by an indicator and an odontometer.

The total measurement of the indicator needle was recorded for every tooth interval and not for any sudden jump during that tooth interval. This of course recorded only acceleration or deceleration and not the type of error, so that such readings were negative, and yet served the purpose of an approximate check.

The odontometer was used to measure the maximum tooth error as indicated on the charts, which also served as an approximate check.

Table 2 shows the result of errors for the different runs as determined by indicator, odontometer, and chart diagrams, including a column of "specified or normal errors." The series of hardened and ground steel gears were purposely made with specified errors as shown in the table. In addition, two pairs of commercial cast-iron gears were included in the test.

The column of assumed errors gives values which were used in the calculations. The values of all except three of the combinations were taken directly from the charts: A and C, D, and F showed values on the chart which were materially less than the probable values, as shown by odontometer tests and the loads of the test runs. On all of these three combinations the maximum errors were decelerating errors, which in every case the charting mechanism failed to record. In addition it is possible that the value used for run L is less than the actual maximum error as shown by the charts, because the loads on this run are materially higher than should otherwise be expected.

The Load-Velocity Tests

After the charts showing the errors of the meshing gears had been drawn, this mechanism was disconnected, and the machine arranged for testing the gears at speed. First the initial pressure between the mating teeth was established by means of the torsion of the telescope shafts bearing the gears. The motor and recording mechanism were then started and the speed increased until the breaking of contact between the teeth, due to the speed of the gear, was announced to the observer by the clicking of the telephone receivers.

These observations were repeated for various initial loads, which established the relation between speed and increment loads, as affected by the various factors, of speed, error, elasticity of materials, and other factors which remain to be determined.

Fig. 5 is a diagram showing pressure on gear teeth as affected by momentum at speed. At slow speed the pressure on the teeth is approximately constant, but as the speed increases it rises and falls as shown by the wave line, maintaining a constant mean, until at some definite speed the contact is broken on the tooth of maximum error. The zero load is the reaction of the maximum load on the tooth after the instant of impact. And it is at this instant that the first click is heard on the telephone receivers at that speed. This maximum load is very nearly equal to twice the static or measured load as shown on the diagram. For this load, if the speed is increased the breaking of contact increases until it sounds like miniature thunder or static in a wireless set. With a balanced condition, that is, when the increment load is approximately equal to the static load, the influence of breaking in contact is transmitted through the flexibility of the pinion shaft from the test gears to the master gears at a slightly greater speed.

The results of the tests so far performed are plotted in Figs. 6 to 17.

The amount of the error rather than its nature seemed to be the determining factor in the loads required to keep the teeth in contact.

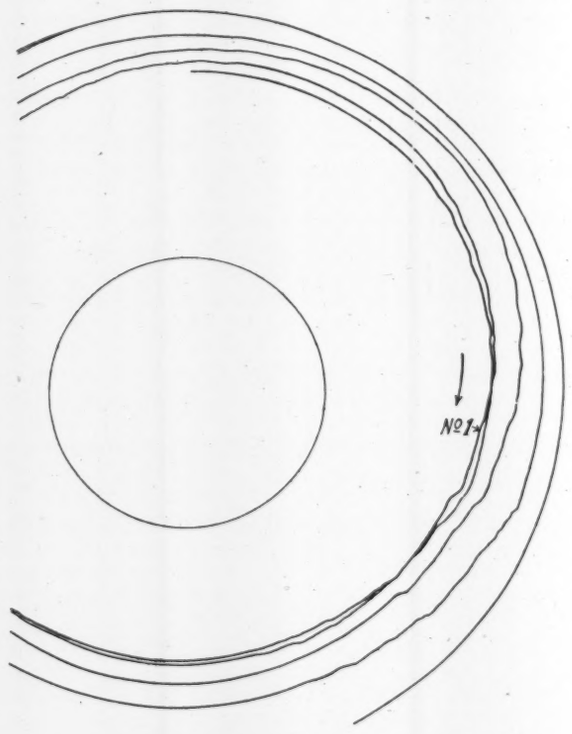

Fig. 4 Composite Errors of Master Gear and Pinion—Chart N

The results of the tests on the steel gears and pinions reduced to a common error of 0.001 in. are plotted on Fig. 18.

A tentative formula has been derived and the curve shown on each of the diagrams represents the values as determined from this tentative formula.

Discussion follows which shows the derivation of the tentative formula.

A comparison of the results with those of Professor Marx is plotted on Fig. 19. A further analysis of these and Professor Marx's tests which shows the conditions that would exist with a constant transmitted load at all velocities is also plotted on Fig. 20.

Effect of Velocity on Load

The tests on the Lewis machine indicate that the effect of the velocity on the loads required to keep the teeth of the gears in contact is approximately the square root of the cube of the velocity. At low speeds the loads are greater, while at higher speeds the loads are less than those proportional to $V^{3/2}$.

An equation that approximates very closely the loads given by the tests is of the following form:

$$L = \frac{V^{3/2}}{1200 + V} C_1 \quad \ldots\ldots [1]$$

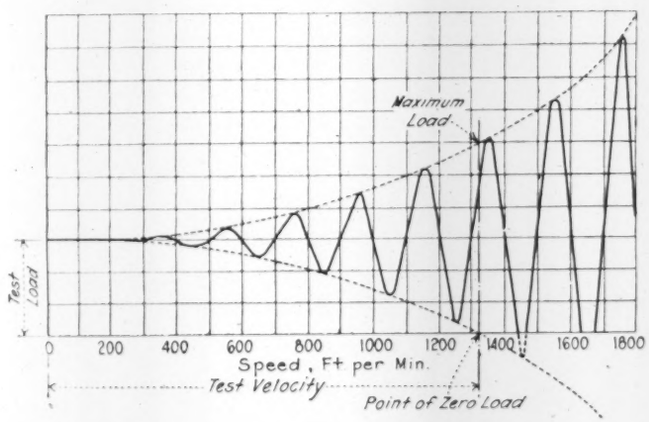

FIG. 5 PRESSURE ON GEAR TEETH AS AFFECTED BY MOMENTUM AT SPEED

where L = load on testing machine in pounds
V = velocity in feet per minute
C_1 = constant for any given pair of gears, depending upon error, material, pitch, tooth form, etc.

Effect of Errors

The present tests seem to indicate that the effect of errors in the tooth profile or spacing is directly proportional to the extent of the error regardless of its nature.

Eccentricity seems to have no appreciable effect on the loads required to hold the teeth in contact. The pinion used for run M was badly eccentric, as may be seen from the chart, but the loads on this test were no greater than normal.

All of the gears tested were of 3 diametral pitch. These same errors on gears of different pitch would have a different effect. The relationship between the errors and the pitch should be determined by further tests. In the meantime this relationship will be established by analysis when we consider the effect of tooth form and pitch.

Introducing this error factor, e, the equation becomes

$$L = \frac{V^{3/2}e}{1200 + V} C_2 \quad \ldots\ldots [2]$$

where C_2 = constant for any given pair of gears, depending upon material, pitch, tooth form, etc.

Effect of Elasticity of Material

From the present incomplete series of tests, it would seem that the loads developed by a given error varied directly as the modulus of elasticity of the material of which the gears are made. Three combinations of materials were tested: steel with steel; cast iron with steel; and cast iron with cast iron.

Another series of tests should be made to test for this factor, using as many different combinations of materials as possible. The

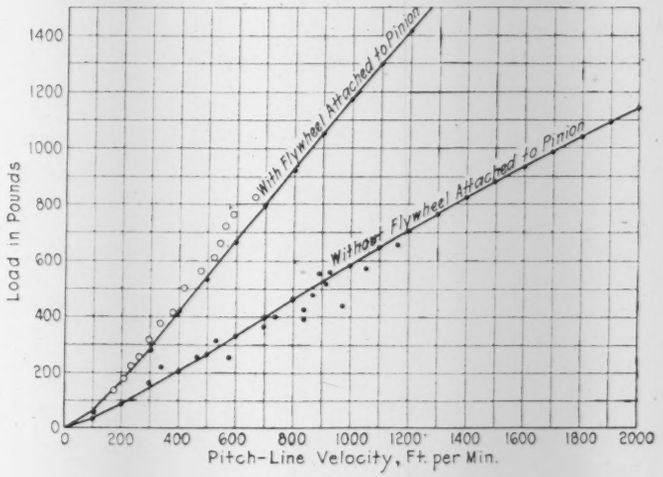

FIG. 6 RESULTS OF RUN B

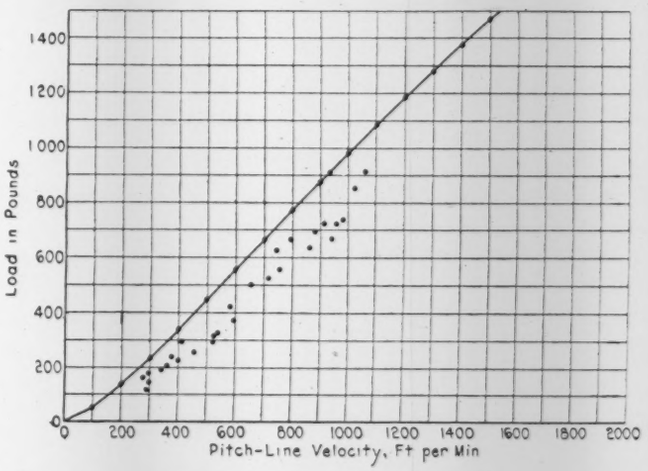

FIG. 7 RESULTS OF RUNS A AND C

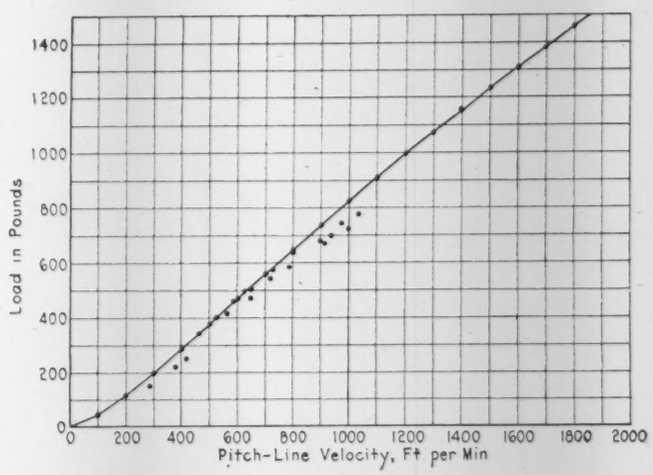

FIG. 8 RESULTS OF RUN D

Newark Gear Company has donated a series of 3 diametral pitch gears of cast iron, steel, bronze, aluminum, and composition or non-metallic materials for this purpose.

Introducing the modulus of elasticity, the equation becomes:

$$L = \frac{V^{3/2}e}{\left(\frac{1}{E_1} + \frac{1}{E_2}\right)(1200 + V)} C_3 \quad \ldots\ldots [3]$$

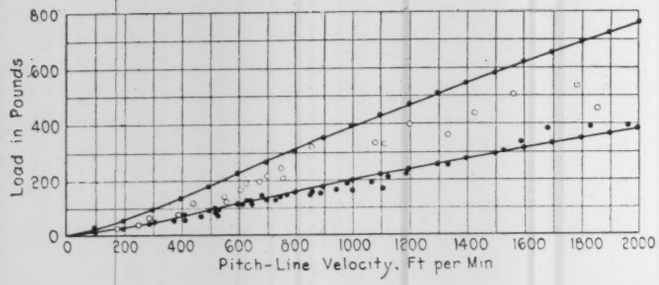

FIG. 9 RESULTS OF RUN E

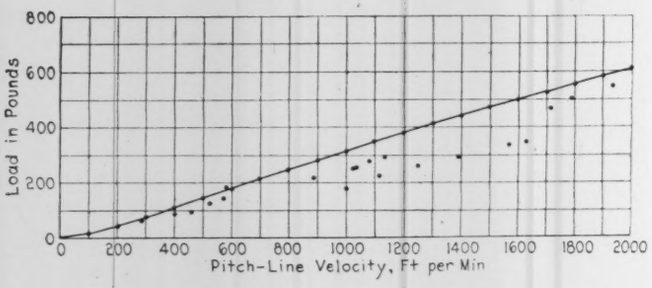

FIG. 10 RESULTS OF RUN F

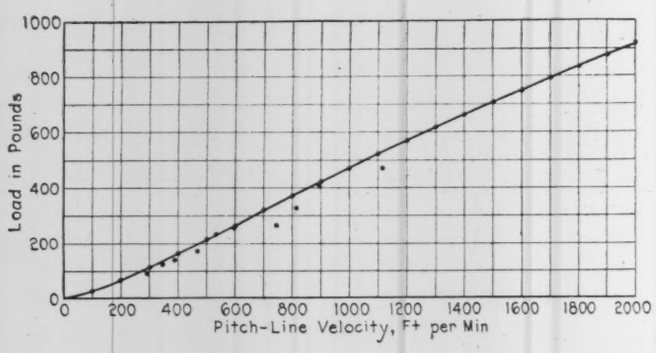

FIG. 11 RESULTS OF RUN G

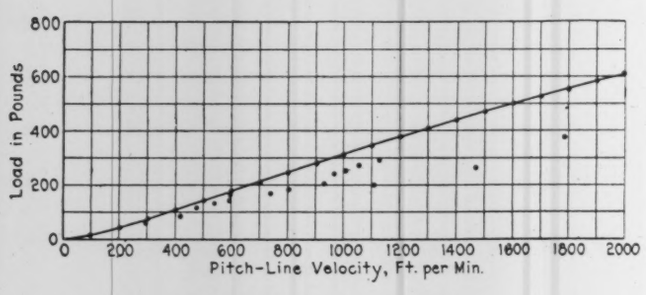

FIG. 12 RESULTS OF RUN H

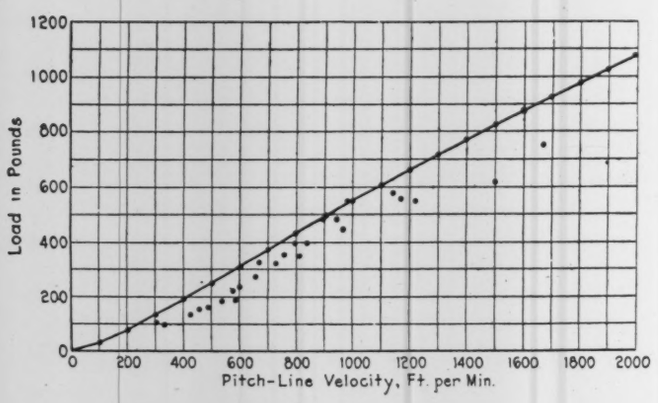

FIG. 13 RESULTS OF RUN I

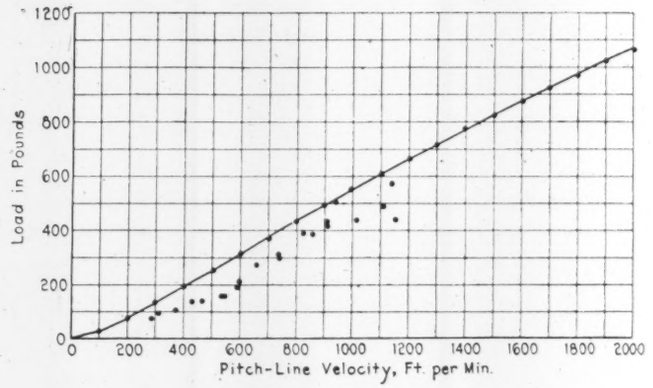

FIG. 14 RESULTS OF RUN J

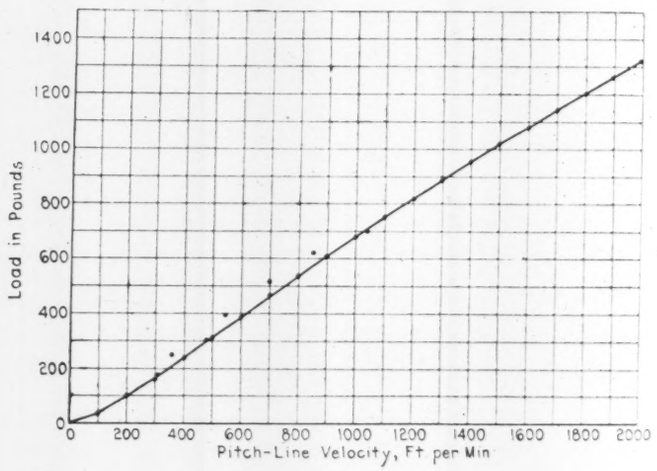

FIG. 15 RESULTS OF RUN K

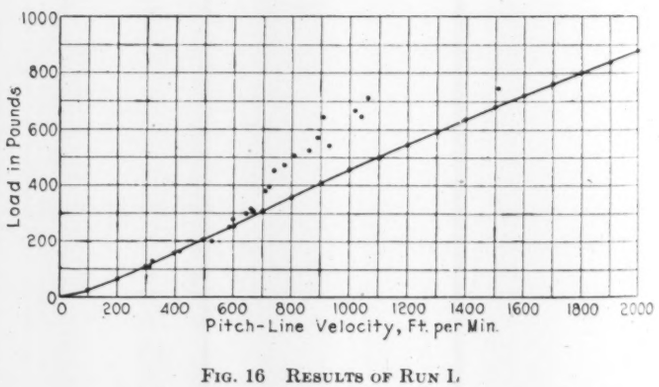

FIG. 16 RESULTS OF RUN L

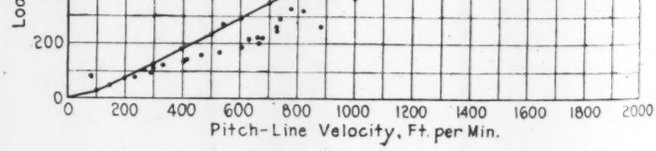

FIG. 17 RESULTS OF RUN M

where C_3 = constant for any given pair of gears, depending upon pitch, tooth form, etc.

EFFECT OF TOOTH FORM

As the loads seem to vary in direct proportion to the modulus of elasticity of the materials, it would be logical to expect them also to vary in direct proportion to the elasticity of the tooth forms. In the present series of tests, all except one pair of the test gears had the same tooth form. The results on the one pair of gears with a weaker tooth form indicated that the foregoing assumption was true. Further tests should be made, however, to definitely establish the effect of this factor.

It is possible that a tooth-form factor for stiffness should be

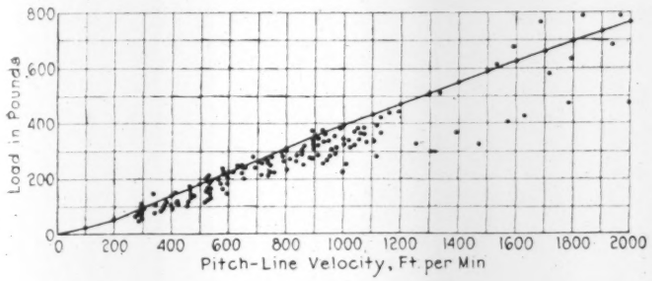

FIG. 18 RESULTS OF RUNS A TO H, INCLUSIVE; EFFECT OF ERRORS REDUCED TO ERROR OF 0.001 IN.

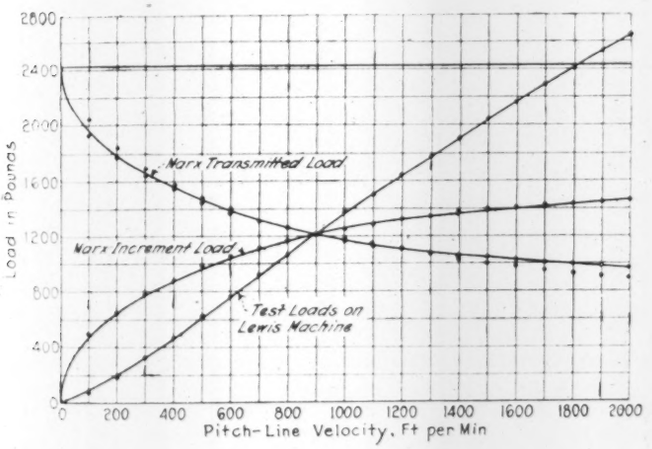

FIG. 19 COMPARISON WITH MARX'S TESTS

established, but this would only be justified if the results of further tests showed it to be necessary. For the present, the Lewis tooth-form strength factor y will be used as a measure of the stiffness. A great degree of accuracy in this factor cannot be hoped for because of the uncertainty of the actual point on the tooth profile where the maximum load is applied—an uncertainty that will always exist on commercial gears.

The use of the factor y also involves the use of the circular pitch of the gears. A change in pitch will have two effects: first, a change in the rigidity of the tooth, and, second, an increase in the number of tooth contacts per unit of time. This second effect is similar to a corresponding change in pitch-line velocity.

The tooth form and pitch factor to be introduced into the equation would thus become

$$\frac{py}{p^{3/2}} = \frac{y}{\sqrt{p}}$$

where p = circular pitch in inches.

As both the stiffness of the gear and that of the pinion should be considered, this factor would become

$$\frac{y_1 y_2}{(y_1 + y_2)\sqrt{p}}$$

Introducing this factor into the equation, we have

$$L = \frac{V^{3/2}\, e y_1 y_2}{\sqrt{p}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)(1200 + V)(y_1 + y_2)} C_4 \ldots\ldots [4]$$

Solving for C_4 we find that it approximates 0.030, which is very close to $1/g$ where g = acceleration due to gravity = 32.2. Whence for the final equation we have

$$L = \frac{V^{3/2} e y_1 y_2 f}{g\sqrt{p}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)(1200 + V)(y_1 + y_2)} \ldots\ldots [5]$$

EFFECT OF MASSES

The first tests were run with two split cast-iron flywheels bolted on the pinion shaft. When one of these flywheels was removed, no appreciable difference in the loads required to keep the teeth in contact was evident. Probably the elasticity of the pinion shaft between these flywheels and the pinion dampened out prac-

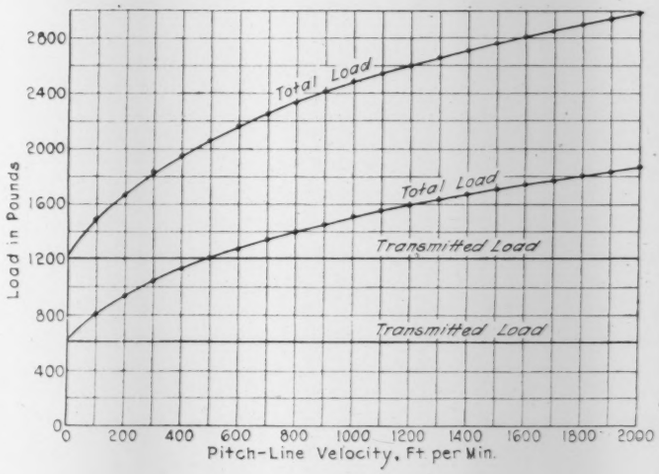

FIG. 20 UNIFORM TRANSMITTED LOADS

tically all of the effects of the difference in mass. The pinion shaft is about three inches in diameter, and the nearest flywheel was about nine inches from the pinion.

Another flywheel, weighing 230 lb., was then attached directly to the pinion. In this case the loads were about double those without this flywheel.

This would seem to indicate that when the revolving masses driven by the gears are an appreciable distance from the gears, the effects of these masses can be ignored; but if they are mounted very close to the gears, they must be considered. Further tests are desirable to determine the effects of masses mounted close to the gears.

EFFECT OF LOADS

It has heretofore been assumed that the velocity ratio for any pair of gears at any given speed was constant, regardless of the intensity of the load. Or in other words, the ratio between the transmitted load and the increment load was constant.

All previous tests on the gear teeth have been made by running cast-iron gears to destruction. It has been assumed that the velocity ratios so obtained were substantially correct for any materials.

Questions have been raised on the accuracy of such tests to destruction, because in every case these tests have shown a rapid falling off in strength at the low speeds, or a rapid increase in increment loads at low speeds, with a constantly reducing rate of change as the speeds increased. It has been pointed out that with rigid materials the increment loads should increase as the square of the velocity, which would give a materially different form of curve from that obtained in tests. With elastic materials the effect of velocity would be modified to some extent, but it did not seem possible to reconcile this theoretical conception with the test results.

As a matter of fact, both the tests and the theoretical conception can be reconciled when the effects of the intensity of the tooth loads are considered.

Referring to Professor Marx's tests, Fig. 19 is shown which also includes a curve of the loads that would be obtained on the Lewis machine when the load at 900 ft. per min. (Marx velocity ratio = 0.500) is the same as that obtained by Marx. In the present tests the weighed load on the machine represents both the transmitted load W and the increment load I. This chart shows that when the transmitted load is increased four times the increment load is doubled, or that the increment load varies as the square root of the transmitted load. Thus we have

$$W_s = W + \sqrt{WL}$$

where W_s = equivalent static load in pounds
W = transmitted load in pounds
L = balanced condition where the transmitted and increment loads are equal.

Conclusion

The general conception is that the velocity factor for a given pair of gears is constant, regardless of intensity of load. All previous tests have been made with cast-iron gears run to destruction. If on these tests at a given velocity the gear failed with a load of one-half its static strength, it was assumed that at any other load the equivalent static load would be double the transmitted load.

Such tests invariably showed a rapid fall of strength at low velocity and a lesser reduction as the velocity increased. This seems contrary to the conception that the additional or increment loads were caused by variations in velocity due to errors, as with rigid solids. Such loads should vary as the square of the velocity. It was admitted that the elasticity of the material would tend to reduce the effect of these errors, but it was questioned whether or not this effect would be sufficient to account for the big difference in the shape of the curve as determined by breaking tests and the curve representing rigid solids.

These tests made it possible to obtain a better idea of the effect of the various factors involved in the strength of gear teeth.

1 The velocity factor is not constant, but depends also on the intensity of the load.

2 Under balanced conditions, such as were maintained in these tests where the additional load was approximately equal to the transmitted load, the curve representing these loads followed in general the shape that would be obtained from rigid solids.

3 The influence of the elastic properties of the material was very apparent and flattened the curve out more and more as the loads and speeds increased, thus counteracting the influence of errors.

4 The influence of error is very pronounced, and seems to be directly proportional to the load required to keep the teeth in contact, thus building up increment loads which are very nearly equal to the transmitted load. By introducing the effect of change in load by using data from Professor Marx's tests, it was possible to reconcile the idea of increment loads as caused by errors with results of previous breaking tests on cast-iron gears.

5 Suggestions for further tests to obtain more definite values for the effects of other factors, such as material, tooth form and pitch.

6 For the same extent of error the increment loads are less for a pair of meshing gears of softer materials, due to the reduction in their modulus of elasticity.

7 The noise of a pair of gears seems to be directly proportional to the extent of error. The noisiest pair of meshing gears, while running was that having the largest error. With same error and different materials the noise is not as great.

8 Run D which showed the maximum indicator error of 0.0043, proved to be the noisiest of the series. In this run the pinion GX3–18–4 had a normal pitch specified error of 0.002 in one tooth only. Thus when running at a 1000-ft.-per-min. pitch-line velocity and static load of 725 lb. it caused distress on the master pinion. This distress showed up on every sixth tooth of the pinion and gear, the master set. The number of teeth being 18 and 48 for pinion and gear and greatest common divisor is 6—hence the symmetry for distress as they meshed in running. It is interesting to note that it was the master set that took the brunt, whereas the pinion with the error remained untouched and smooth. When signs of distress show up, the limit of safe load has been reached and any further running ruins the gear very rapidly.

9 Since the increment loads may be equal to or greater than the static load, and directly proportional to the errors, the error should be kept to a minimum to insure safe stress and durability.

10 The derived tentative formula [4] is cumbersome in its use; however, a table containing various factors involved, based on unit load L/f, where f is the face of the gear in inches, with different combinations of materials, pitch and of factors as well as velocities, will be compiled, which will permit its use with ease.

Hypoid Gears

HYPOID gears are tapered gears with offset axes. In general, they look like spiral bevel gears, which they will in many cases replace. The tooth action combines the rolling action of spiral bevel gears with a percentage of endwise sliding. Both in design and production, hypoid gears have always presented difficulties, and it is stated that up to the present time no gears of this type, with mathematically correct forms of teeth adapted for successful commercial production, have been known.

The chief advantages of the hypoid gears, the characteristics of which are outlined below, are noiseless operation, increased load-carrying capacity, the possibility of high reduction and low numbers of teeth, long life and high efficiency. These advantages are said to have been demonstrated in severe tests of these gears in rear axles of motor cars.

The most striking difference between hypoid and spiral bevel gears is the offset of the pinion axis from the axis of the gear, on the hypoid pair. Another marked difference is the increased size of the hypoid pinion as compared with the bevel pinion. Furthermore, it is to be noted that the tooth inclination on the hypoid gear, or its spiral angle, is small, while the spiral angle of the hypoid pinion is as large as or larger than the spiral angle of the bevel pinion. All of these differences have an influence on the performance and the characteristics of hypoid gears, as outlined below.

The new developments made at the Gleason Works comprise new tooth shapes for hypoid gears as well as new methods of production, and are covered by a series of pending patents. The tooth shape is derived from surfaces of revolution and permits the use of the usual type of rotary cutters for generating the teeth.

Two different types of tooth shape and two corresponding methods of production have been developed. The earlier method consists in forming the gears without generation. A tooth space of the gear is then the exact complement or counterpart of the cutting surface of the tool. According to a general law of gearing, the tooth surfaces of one member of any pair of gears may be assumed at will, and the tooth surfaces of the other member are then determined or generated to suit the tooth surfaces of the first member. This is done by setting a cutter on a generating machine in such a way that it covers and represents a tooth side of the hypoid gear, and by providing the same relative motion between the cutter and the pinion blank which would exist when the pinion meshes with its mating gear.

The hypoid non-generated gears can be cut on the machine now used for roughing spiral bevel gears, but, to obtain the quality and accuracy required for finished gears, machines of higher grade than the gear roughers would be required. A later method permits cutting the gears on the known standard generators, exactly like spiral bevel gears. The pinions are cut in either case on a machine which differs from the standard generators by permitting additional adjustments. In the latter case, the generating roll is affected, as if the pinion would roll and slide on a crown gear, while axis is offset from the axis of the pinion.

The axis of the pinion is offset from the axis of the gear by an amount which varies with the diameter and with the ratio. Standard amounts of offset of $1\frac{1}{2}$ in., 2 in., $2\frac{1}{2}$ in., 3 in., and $3\frac{1}{2}$ in. are in use.

The direction of offset determines the hand of the spiral. In rear-axle design a pinion below center will have a left-hand spiral while a pinion above center will have a right-hand spiral.

Objections to the above center position of the pinion, as far as end thrust is concerned, can be entirely eliminated by making the bearing and housing arrangements sufficiently rigid to prevent the pinion from wedging into the gear. (From paper by Arthur L. Stewart and Ernest Wildhaber, *Iron Age*, vol. 118, no. 2, July 8, 1926, p. 84.)

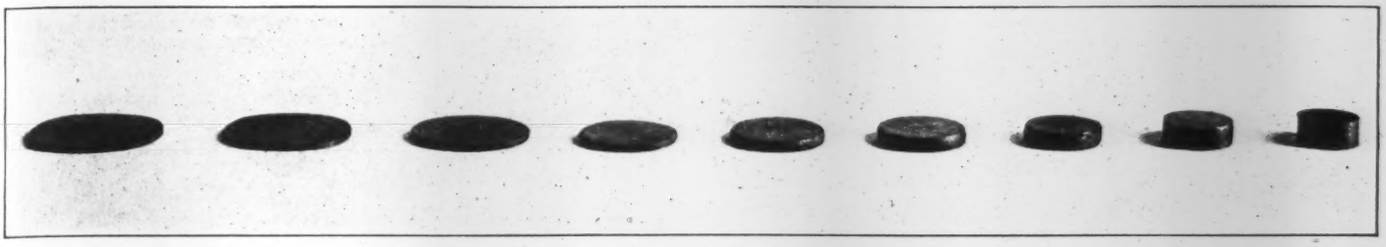

FIG. 1 COPPER SLUGS COMPRESSED COLD BY VARIOUS AMOUNTS

Cold-Press Finishing of Metal

BY E. V. CRANE,[1] BROOKLYN, N. Y.

COINING presses are not new. Their history goes well back into the last century. Nor is the cold squeezing of the softer metals new. The early machines of this type were developed for, and took their name from the coining of the precious and semi-precious metals for money and medals. The development of widespread competitive marketing necessitated making use of every short cut in quantity production. The possibility of squeezing a shape or a surface from a rough blank or forging to accurate size, cold, and in a small fraction of a minute is naturally attractive, especially on interchangeable work.

Squeezing steel cold is practically the severest of all press-working operations. Depending upon the shape of the job, the design of the dies, etc., the working pressure may climb to one hundred or more tons per square inch on the surface of the product. Coinage methods have shown the way but naturally there has been time and money lost pending the development of alloy die steels that would stand up and the adaptation of the presses to the severer conditions.

We have built several hundred knuckle-joint-type power presses, the majority of which are now in use on this type of work. But for every one in use now there will probably be ten, a couple of decades from now. Quite a few of these machines are in use in this district, but with only about two exceptions, the New England manufacturers to whom we wrote in our efforts to broaden this paper, considered their cold swaging work in the light of a secret process. Such is not the case and cannot be if manufacturing methods are to progress.

For present purposes, cold squeezing or the working of metal in compression in power presses may be divided into four classifications covering common applications. These are arranged approximately according to their severity although the conditions of individual cases may upset this order entirely.

Usually the least severe comparatively is the mere sizing, flattening, or surfacing of forgings, stampings, or castings, accompanied by very little reduction of thickness (see Fig. 5).

Second is swaging, cold forging, or upsetting where a suitable blank or slug is forced into a desired shape to save machining operations, usually involving considerable reduction of thickness in some portions but characterized by considerable unrestricted freedom of flow.

Third is coining, stamping, or embossing where the metal, pretty well confined and usually in comparatively thin sections, may be forced to flow to fill the shape and profile of the dies.

Fourth is extrusion where the metal is forced to flow plasticly through an orifice of whatever shape, being otherwise confined. This is quite limited as to metals and applications and will be considered as outside the scope of this paper.

In all four classes the metal must be squeezed beyond its elastic limit in compression. The deformation takes place as sliding along the slip planes of the crystals, resulting where the deformation is considerable in appreciable reduction of grain size and hardening of the material. If it is desirable, the original grain size can again be approximated by suitable annealing after the squeezing operation. A few preliminary results from experiments on cold squeezing performed in the Bliss testing laboratories illustrate some of the points.

PRESSURES IN DEFORMING METAL

Fig. 1 shows a series of copper slugs compressed cold by varying amounts. All were cut to identical dimensions from extruded rod as illustrated by the slug at the right. They were squeezed between hardened and ground tool-steel blocks in Olsen testing machines step by step down to about twenty per cent of their original height. The elastic limit was apparent at very close to 44,000 lb. per sq. in. or 22 tons per sq. in. checking closely with values commonly given for the compressive strength of copper and rising evenly to over 40 tons per sq. in. for the slug at the left. A curve plotted from over a hundred readings is shown in Fig. 2. The rise in pressure per square inch is accounted for, in part, by the hardening of the material which varied from sclerescope No. 73 to No. 93 for the slugs shown in Fig. 1, and to a greater extent to the shape of the slugs, the relation of diameter to thickness. If a large thin disk be divided into concentric rings, Fig. 3, and the volume and area relations are considered, it will be clear that in squeezing the disk thinner, the outer rings tend to act as tension bands resisting the expansion of the inner rings. The result is a pyramiding of the pressure toward the center amounting on thin disks to a considerable factor. It may be noted that the slug at the left of Fig. 1 shows the beginnings of tensile fractures around its periphery and that the higher points on the curve (Fig. 2) show the effect of this failure about the edge in a deviation from the true curve. Isolated examples of compression of copper disks approximately $1^1/_2$ in. diameter by $^3/_{16}$ in. thick have shown pressures exceeding 150 tons per sq. in. Here the disks showed appreciably greater thickness toward the center than at the edges due to yielding of the die steels under the extreme pressure. This pyramiding of pressure is worthy of note as quite a few swaging applications, including some of the samples illustrated, show its effect.

Round steel slugs tested in a manner similar to that employed on the copper slugs, though not carried as far, showed similar tendencies. A 0.12-carbon slug was compressed to half its original length, the pressure rising in a nearly straight line from about 45,000 lb. per sq. in. through 74,000 lb. per sq. in. (37 tons per sq. in.) with an appreciable hardening of the slug.

Fig. 4, curves from an older test on flattening round bars show pressures up to about 135 tons per sq. in. and 190 tons per sq. in. In all of these tests it should be noted that the dies have been merely flat plates, the metal being free to flow in other directions. Other cases will be referred to later, though it may be noted here, that Unwin places the "pressure of fluidity" for copper at 54,000 lb. per sq. in. and for mild steel at 112,000 lb. per sq. in. However, as the tests indicate, a definite value for an abstract case is difficult to give.

SIZING OF FORGINGS

Under the first classification, sizing work, where the metal is ordinarily not displaced appreciably and such flow as occurs is not restricted, the most general application and the one receiving most attention at this time is sizing drop forgings. Fig. 5 shows a random selection of drop-forged automobile parts on which flats,

[1] Staff Engineer, E. W. Bliss Co.

For presentation at the New Haven Machine Tool Exhibition, Mason Laboratory, Yale University, New Haven, Conn., September 7–10, 1926.

angles, or boss faces have been squeezed to size in knuckle-joint-type presses. The object of course is to squeeze the boss surfaces of the unfinished forgings accurately to size so that the usual milling or grinding operations are eliminated. The division of the General Motors Corporation which furnished these samples advised that their usual tolerance for such work was ±0.001 in. This is fairly

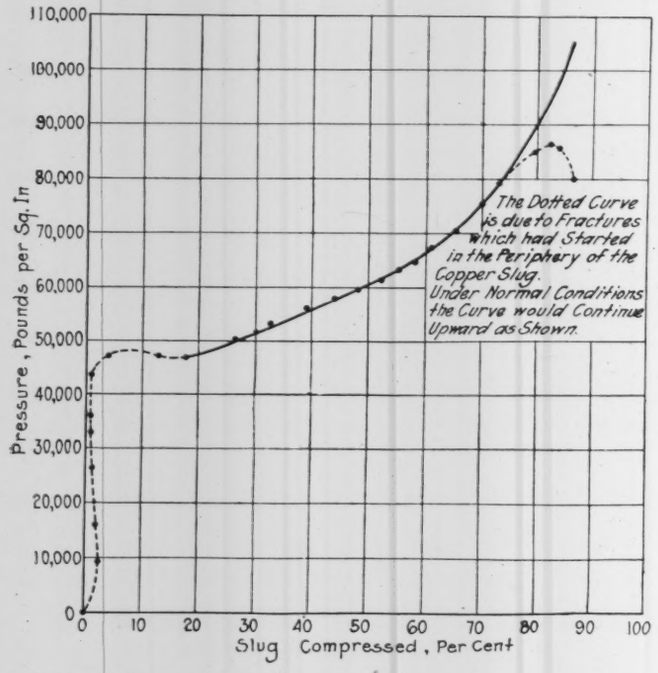
Pressure, Pounds per Sq. In
Slug Compressed, Per Cent
The Dotted Curve is due to Fractures which had Started in the Periphery of the Copper Slug. Under Normal Conditions the Curve would Continue Upward as Shown.

FIG. 2 PRESSURE CURVE FOR COMPRESSION OF COPPER SLUGS

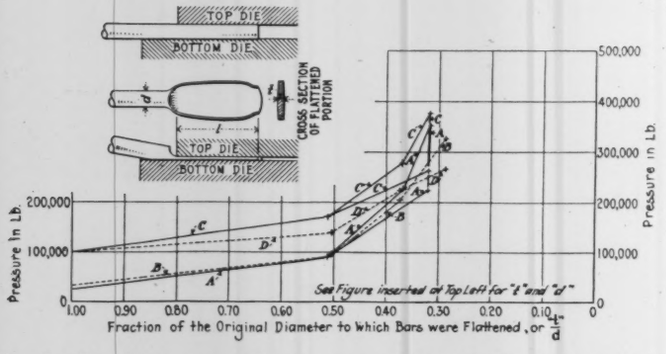
TOP DIE
BOTTOM DIE
CROSS SECTION OF FLATTENED PORTION
Pressure in Lb.
See Figure inserted at Top Left for "t" and "d"
Fraction of the Original Diameter to Which Bars were Flattened, or t/d

FIG. 4 COLD FLATTENING TESTS OF THE ENDS OF CYLINDRICAL STEEL BARS

(The lines A and B lead through points showing pressures per inch of "*l*" (see "*l*" in inserted figure). Line A is for ⅜-in. bars and line B for ½-in. bars. The lines C and D lead through points showing pressures per sq. in. of top surface of flattened portion of bar. Line C is for ⅜-in. bars and line D is for ½-in. bars.)

representative and is closer than tolerances ordinarily allowed on the milling of similar parts.

The dies for sizing bosses are usually plain blocks with comparatively large surface area so that the forging need not be located with especial care. Under favorable conditions one operator and one press can size cold four to eight or ten times as many parts as can be milled or ground by one operator with a suitably rigged machine, and where the quantity is sufficient and the shape of the part is adaptable, a magazine-type push feed, Fig. 6, can be applied to the press and the production rate increased by another twenty-five or fifty per cent.

The forging to be sized, may of course have a number of different finish surfaces at varying levels which are taken care of by corresponding steps in the die. The relative heights of such bosses on the sized forging are found more reliable than they would be if the piece had been machined. Sizing of forgings is not limited to the flat surfaces of bosses. Rounds, bevels, sides, tapers, flanges, even the inner surfaces of punched holes can be sized smooth and accurate. One thing to watch especially in sizing steel is to have ample space for the free flow of the metal, not to restrict it so that the pressure builds up. A certain amount of restriction is possible when necessary but it necessitates greater care in construction of the die and the life of the steels is proportionately reduced. It is of interest to note that forging a depression in the center of the bosses, as in the case of the shift levers at the sides in Fig. 5, reduces the area, prevents pyramiding of pressure at the center which is hard on the dies in the case of a large-area squeeze, and localizes any deformation at the ends of the boss.

The usual finish allowance for work of this character is $^1/_{32}$ in. and in some cases $^1/_{16}$ in. More can be allowed but is not required. It may be pointed out that if the rough forgings vary considerably the pressure which the press is called upon to exert will vary as indicated by the percentage compression curve, Fig. 2. Accordingly if the press is not amply heavy and of rigid construction it will spring in proportion to the load giving a possible variation beyond the tolerance. A press of the built-up type, Fig. 16, with the steel tie rods which tie the bed and crown *properly shrunk*, will not spring in the side members up to its full rated load. If then the bed and crown are amply stiff and the static load requirement of the job is well within the rated capacity of the machine, a considerable variation in the forgings should be absorbed without exceeding the tolerance. It is best, however, to figure the pressure requirement for this sizing work (close tolerance, small compression, free flow) at about 60 to 80 tons per sq. in. of surface to be squeezed. It may be noted that

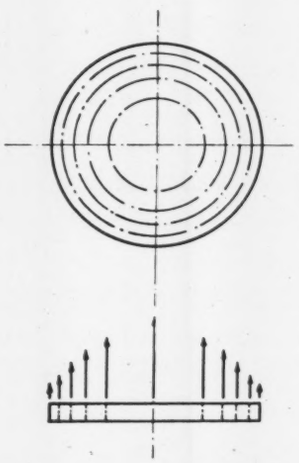

FIG. 3 VOLUME AND AREA RELATIONS OF A DISK IN COMPRESSION

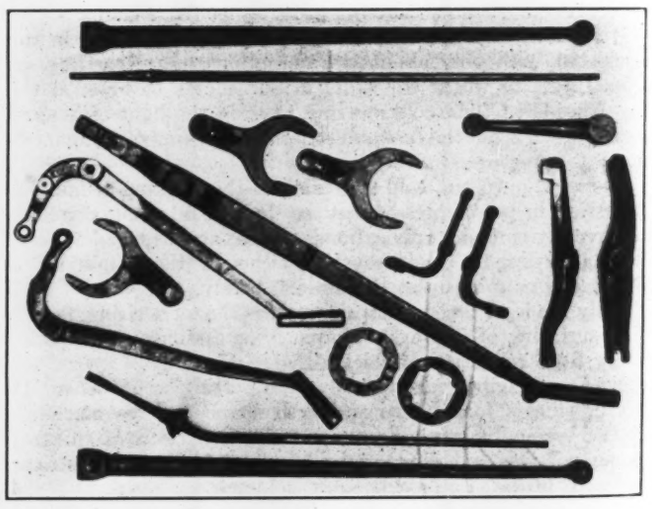

FIG. 5 DROP-FORGED AUTOMOBILE PARTS ON WHICH FLATS, ANGLES OR BOSS FACES HAVE BEEN SQUEEZED TO SIZE IN KNUCKLE-JOINT PRESSES

400-, 600-, and 800-ton capacity presses seem to have proved to be the most satisfactory for the general run of automobile forgings.

Those who are especially interested in this phase of cold swaging will find a very complete paper on it by A. R. Kelso, published in *Journal* of the Society of Automotive Engineers, November, 1924.

SIZING OF CASTINGS

Another line of work under the first classification but not as common as the above is the sizing of castings. Many brass, bronze, aluminum, and alloy castings and some steel and malleable castings lend themselves to cold sizing and surfacing methods. The bosses on the two L-shaped cast brake levers at the bottom in Fig. 5 are squeezed to size. In such cases, the same general comments apply as have been made on the sizing of drop forgings except that the

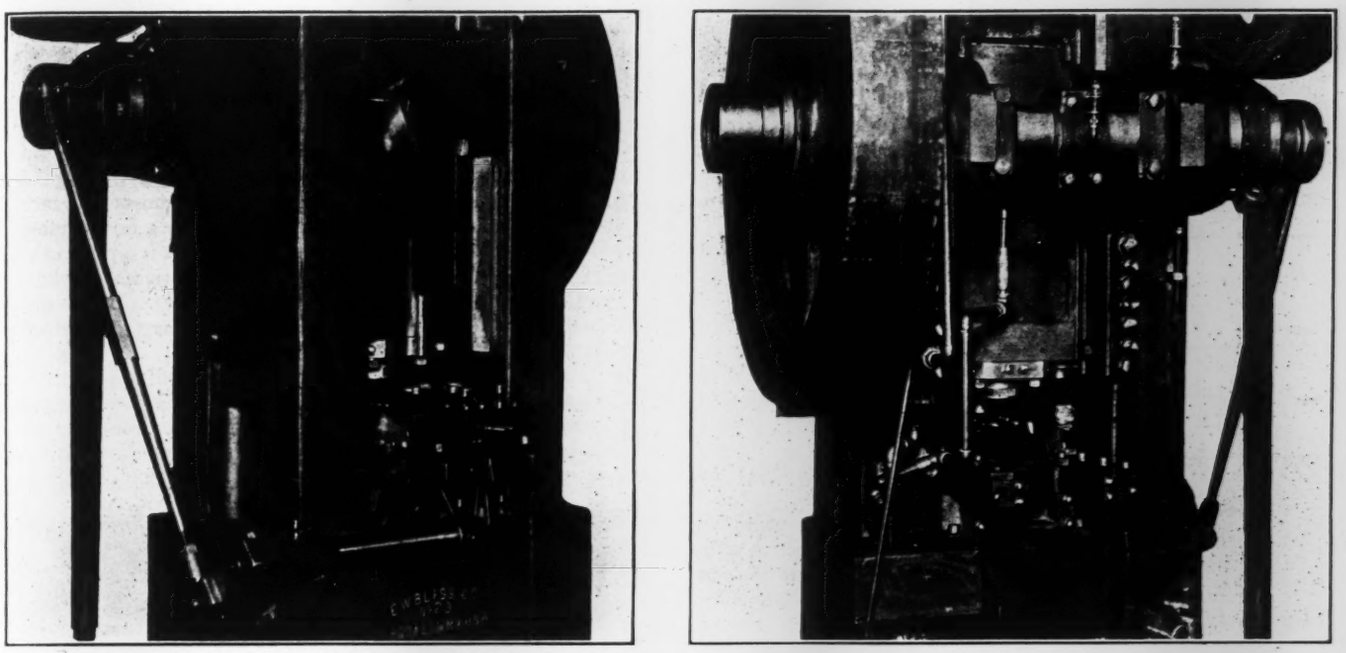

FIG. 6 KNUCKLE-JOINT PRESS WITH MAGAZINE-TYPE PUSH FEED

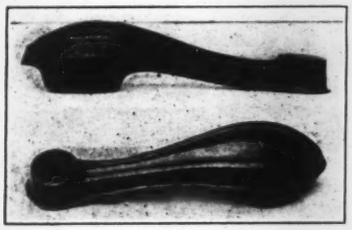

FIG. 7 CAST HANDLE COINED FOR NICKEL PLATING

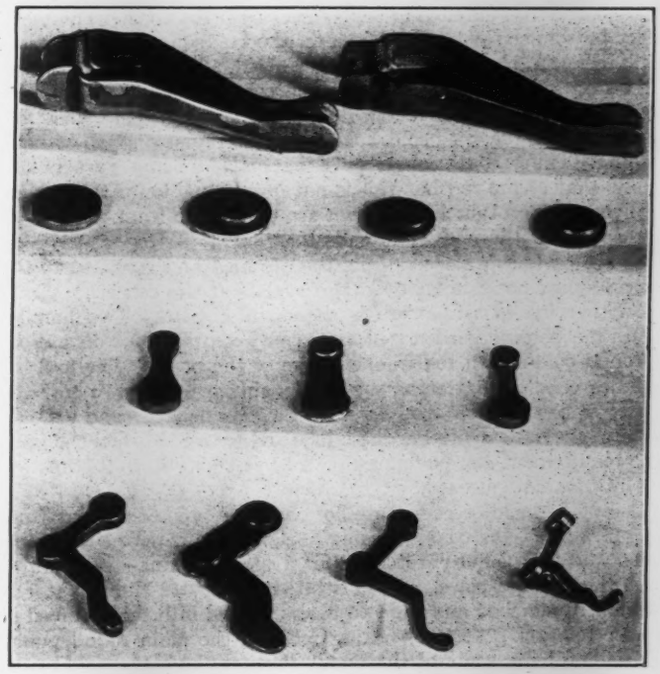

FIG. 8 SQUEEZING AND SIZING JOBS

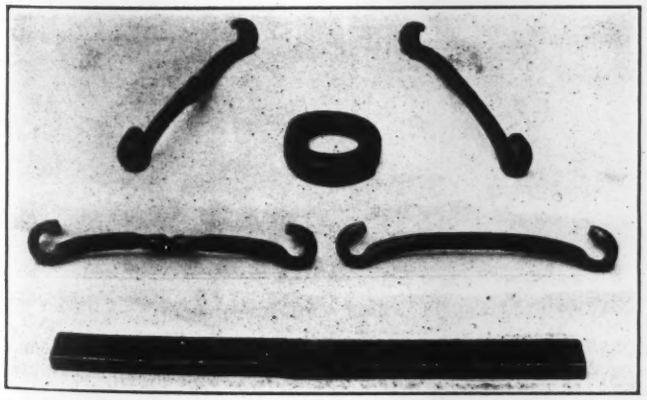

FIG. 9 ROUGH PARTS MADE FROM HOT-ROLLED BAR OR ROD EMBODYING COLD-SQUEEZING OPERATIONS

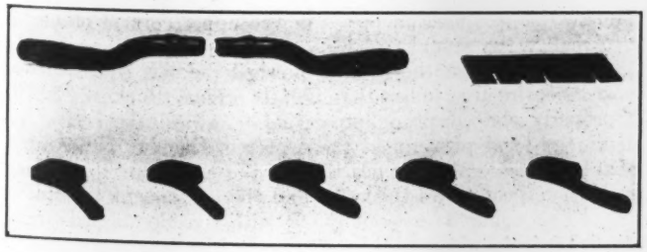

FIG. 10 EXAMPLES OF COLD SWAGING

FIG. 11 PRESS RIVETING BY COLD SQUEEZING

pressure requirements run lower for the softer metals and somewhat higher for the steel and malleable castings. In the case of brass there is an increasing tendency to go to hot press forgings for many shapes, the accuracy of which can be held very nearly as close as what could be obtained by cold sizing.

Another application of coining methods to castings is found in the preparation of some classes of hardware for plating. The surface of the casting can be coined sufficiently smooth to eliminate or greatly reduce the preparatory hand-work and buffing. A cast handle shown in Fig. 7 is coined in this manner for nickel plating, a fin of surplus metal being forced out which must be shaved off later. The pressure required for such a job is likely to run about 90 tons per sq. in. of projected area.

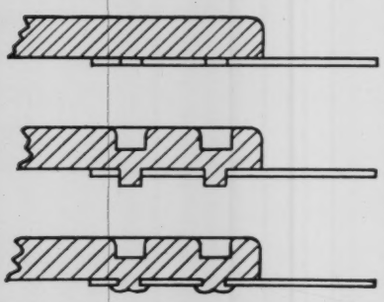

FIG. 12 RIVETING BY COLD SQUEEZING

Between the first and second classification are a variety of squeezing and sizing jobs on which either the pressure required is not great or accuracy is not essential. At the top of Fig. 8 are shown two samples of an 11-gage ($^1/_8$-in.) steel lever. This lever is blanked, formed, and then placed in a sizing die which sets down the sides thickening them slightly, squaring up the corner radius somewhat, and squaring off the edges.

In Fig. 9 are shown a number of rough parts made from hot-rolled bar or rod employing cold-squeezing operations. The piece at the bottom shows two wedges after the first squeeze which forms the taper. A sharper edged squeeze will all but complete the edges and sever them. Above in the center is a link made of a round rod formed with the two ends overlapping. The ends are then squeezed together cold, making an amply strong job without welding. The cross-sectional area of that end of the link is practically double that of the rod but this is not objectionable for the service involved. The other links shown are also made from round rod. The space for the ends, two ends together, is flattened; the long links are parted in a second die, then formed and returned to the knuckle-joint press to coin the two eyelets at the center.

Another two-at-a-time job is shown in Fig. 10. Two handles for an inexpensive garden tool are formed complete out of a piece of round cold-drawn rod in one stroke of the press. The forming includes bending, flattening the end leaving two rivets to hold the blade, squeezing two stop ears on the handle end, and parting. Accuracy is not essential in these jobs and for a mild steel the pressure requirement can be figured on a basis about 80 to 90 tons per sq. in. of area squeezed.

It may be of interest to note here another type of riveting from the stock involving cold squeezing. Fig. 11 shows part of an equipment built for the Ford Motor Co. for riveting a spring-steel vibrator to a steel bridge piece. As illustrated in Fig. 12 the vibrator is previously punched and the two pieces are fed into the dial feed in proper relation. At the working position the plain round punch descends forcing some of the metal down through the hole in the vibrator. The die is so arranged, however, that the metal is turned back at edges as it comes through completing the riveting in one stroke of the press. This equipment is operated at the production rate of about sixty assemblies per minute.

COLD FORGING

Distinctly in the second classification of swaging, cold forging, etc., combining great accuracy required and severe working stresses, are a quantity of odd shaped pieces squeezed to shape out of solid metal for parts for adding machines, sewing machines, speedometers, typewriters, electrical equipment, and even toys and novelties. Fig. 13 shows a typical press for this sort of work, a 400-ton knuckle-joint press swaging an adding-machine wheel. Figs. 8, 10, and 14 contain a random collection of samples which illustrate many of the possibilities of this method.

To start with a simple case, across the bottom of Fig. 10 are shown the operations in the production of a medium hard steel cam lever. At the left is the blank, pierced and blanked from strip stock. Next the grip portion of the lever (extending to the front) is squeezed top and bottom as it lies, to round up the corners and end. The third operation is a squeeze which bends and flattens the grip portion to its final shape. Fourth, the cam way is squeezed into the head portion, reducing the metal thickness there by nearly fifty per cent. Fifth, the edge around the head portion is shaved smooth so that the completed lever is ready for plating. Here in five quick press operations including three squeezes, a lever has been turned out which would otherwise require a hot forging and considerable hand-work. Above this at the right is a plate in which flanges have been squeezed to about half the original thickness of the metal, down each side of the three deep slots. Before making the swaging die for this operation it was necessary to machine out the flange-ways with a small end-mill.

At the left, second row up, in Fig. 14 is a small gear with a cam tooth on it, originally built up of two pieces riveted together at considerable expense and with a high scrap loss due to slight inaccuracies in the assembly of the two parts. As shown it is swaged in one operation from the blank shown at the left and trimmed

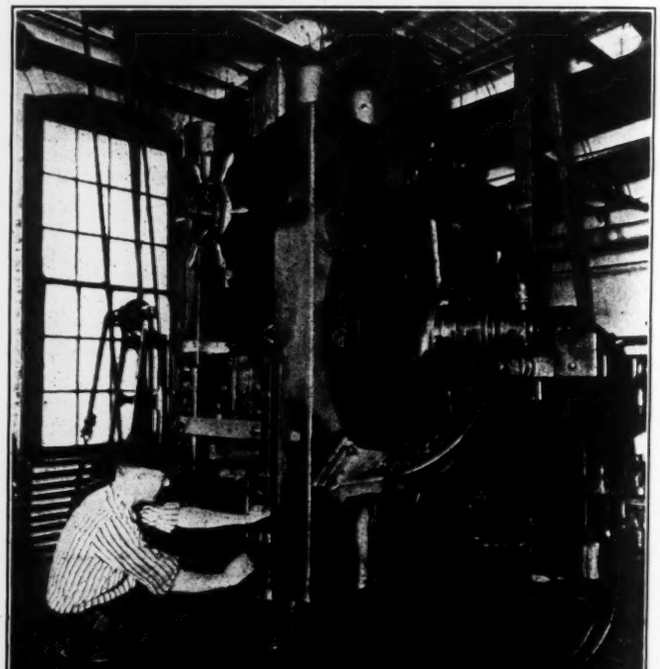

FIG. 13 PRESS FOR COLD FORGING

so that the teeth and cam are in exact relation. The solid one-piece construction proved both better and cheaper than the built-up original. An excellent article describing this and several similar pieces and the dies for a couple of them was published by D. A. Chason in the *American Machinist*, Feb. 22, 1912. Another paper by Mr. Chason without die details, however, was published in MECHANICAL ENGINEERING, September, 1924.

The two simple looking pieces in the upper corners of Fig. 14 are in reality the severest of those shown. The rectangular slug slightly thicker than the desired finished piece is squeezed over almost its entire area to less than half its original thickness, leaving the properly shaped prong projecting. Toward the end of the squeeze the local pressure at the center of the slug builds up to probably twice or three times that at the edges due to the large area with respect to its thickness and the consequent pyramiding effect. This could be materially cut with considerable advantage as to die life if the piece could be squeezed with a hole or slot through the center as a relief. It will be noted that on all of the samples

shown of this class of work, the blank is so designed that as little metal has to be squeezed down as is absolutely necessary for trimming or shaving out the finished piece. Note also that in the pieces which have a boss, hub, or other portion higher than the rest of the piece and left so by squeezing down the metal around it, there is a tendency in the process to drag down the corners and edges of such high parts. To minimize this tendency it is often advisable to use a medium hard stock and if necessary to arrange the dies to strike the high part at the end of the stroke to size it off a little. There may be considerable variation in the pressure required for this work on account of the area and thickness relation. For practical cases, however, with a free flow relief all around it is well to figure 100 tons per sq. in. and higher on the total area squeezed for steel and around 75 tons per sq. in. for copper.

COINING—STAMPING—EMBOSSING

Under the third classification, coining, stamping, embossing, etc., comes a variety of work in which the metal is made to flow comparatively little but is subjected to extremely high pressures to bring out sharp designs or lines or to obtain a very accurate surface. On most of this work the metal is either completely contained in a closed die or practically so, with the result that there is no outlet or relief for the flow of surplus metal. Under such conditions, if care is lacking and oversized stock is used or adjustments are carelessly made, the pressure may build up to several hundred tons per square inch with serious consequences to the equipment.

Fig. 15 shows a number of examples of fairly common coining, stamping, and embossing jobs. The best known application is, of course, government coinage. A report issued some time ago by the Mint showed that the actual pressures required to bring up clear impressions on United States gold, silver, and copper coins varied between 85 and 125 tons per sq. in. *The actual presses used have two or three times the capacity that these figures would indicate* to take care of the difference between static test loads and quick-stroke working loads, and to guard against such overloads as are likely to occur.

The plate shown at the left in Fig. 15 is a severe job in steel being performed on a 1000-ton knuckle-joint press. It is a decorative cover plate coined to resemble a less perfect job previously done by etching with acid. The press is working on a basis of about 110 tons per sq. in. of coined surface with very little margin. At the bottom in Fig. 14 is another steel sample belonging in this class. The plain bar at the left is the blank showing the fractured edge left by shearing. The perforated bar has been squeezed in a closed die coining the top, bottom and sides and destroying practically all traces of the shearing fracture.

Returning to Fig. 15, a fork with its blank, a spoon, and a couple of knife handles indicate what is done with knuckle-joint presses in the manufacture of solid flatware and hollow ware. The fork and spoon coined to shape from solid blanks really come under the head of swaging, while the knife handles made from sheet material are properly embossing work. Other applications of knuckle-joint presses on silverware, jewelry settings, watch cases, etc., are a complete line in themselves.

The two ornamental brass box covers and the copper army button, a 60-ton job, are also embossed in male and female dies without changing the thickness of the metal or flowing it appreciably. Such work can take anywhere from 20 to 200 tons per sq. in. depending upon the setting of the press and the sharpness of the impression desired. A sharp impression depends upon pressure but also to a very great extent upon die construction. Fifty tons can bring up a sharper, better looking job than five hundred if the dies are arranged to pinch only and just where the sharp lines are desired and relieved elsewhere so that the metal can flow a little and pressure be localized. The switch plates at the top of Fig. 15 are a good illustration of this point. One concern making these plates had a great deal of trouble getting the desired sharp line on the face edges, not because not enough pressure was being used, but because too much area was being pinched. On examination of the back of the three-switch plate, which is shown reversed, it may be seen that the pressure was localized in lines less than $^1/_{16}$ in. wide around the button counter-sinks, the edges, and the corners. The pressure requirement figured back to between fifty and seventy five tons per square inch on the area actually squeezed. There are a good many applications of flattening and surfacing where this principle is or could be applied to advantage. This should not be confused with low-pressure flattening jobs performed in light and springy presses where the metal is not squeezed but merely bumped. The very spring of the machine is often the saving grace in such cases, for a real pinch on the metal would jump the load immensely.

While the samples selected show something of what is being done and indicate some of the possibilities in this line of press work, they are naturally not nearly complete. Most of them are designed to do away with some machining, grinding, assembling, or hand operation, and it is well established that with few exceptions, press operations are far cheaper than any of these.

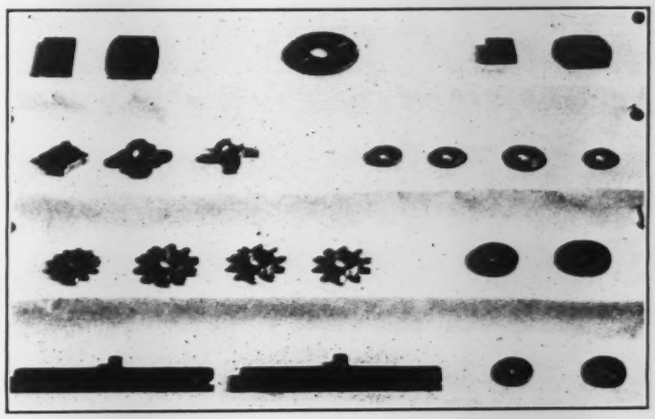

FIG. 14 EXAMPLES OF COLD SWAGING

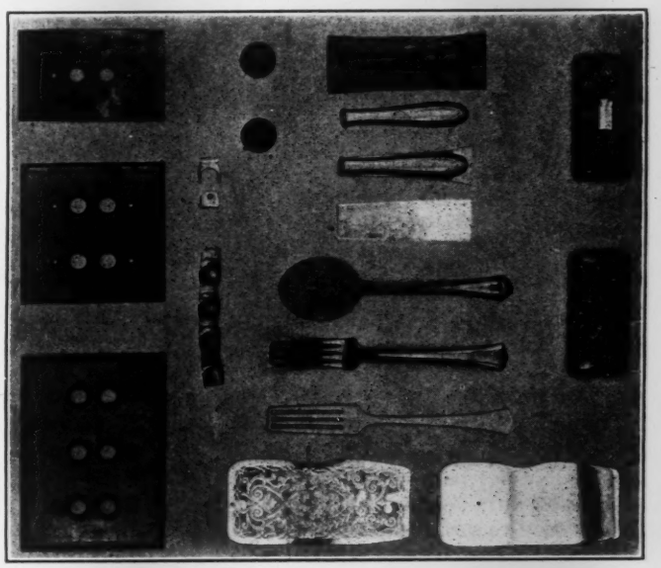

FIG. 15 EXAMPLES OF COINING, STAMPING, AND EMBOSSING

MATERIAL FOR DIES

Difficulties experienced thus far with this type of work have been largely in the selection of too light a press, improper planning of the operations so that high local pressures cut down the life of the dies, and with steel for the dies themselves. It is a dangerous thing to recommend steels for this sort of work for so much depends upon proper and exact hardening and drawing, and steel-treating methods vary extremely. We have, however, compiled a list of steels for

TABLE 1

	Trade Name					
	Styrian Purple Lable	Crucible H.Y.C.C.	Colonial Special	Colonial 7D	Vasco K	? (A. R. Kelso)
Carbon	1.08	2.25	0.96–1.05	0.85–0.95	0.76–0.85	0.67–0.80
Manganese	0.21	0.30	0.20–0.30	0.20–0.30	0.10–0.30	0.20–0.30
Silicon	0.22	0.40	0.10–0.20	0.10–0.20	0.20–0.35	0.05
Vanadium	...	0.20	.....	0.18–0.20	0.15–0.25	trace
Chromium	...	12.00	.....	.....	0.70–0.90	0.09
Phosphorus	0.002	0.03–	0.015–	0.015–	0.03–	0.010
Sulphur	0.009	0.03–	0.025–	0.020–	0.03–	0.021

cold-swaging dies, Table 1, to which others, possibly better, should be added, and which we offer only on the ground that successful users of these methods have found them satisfactory. The variety in their analyses indicates how varied are the opinions on what is best.

Under normal conditions the life of the tools on this sort of work appears to vary around twenty to forty thousand pieces per grinding. It has been found to prolong the life of some steels materially, to redraw them, at the redressing periods, to or slightly above their original drawing temperatures. This is apparently accounted for by a tendency to surface crystalization, under the severe repeated stresses, that is, a formation and spreading of minute fatigue fractures between the crystals, a condition which is corrected by the annealing. Another recent "stunt" found to prolong the life of simple shaped dies is to electroplate a thin, hard, smooth layer of chromium, say 0.001 or 0.002 in. over the die steels. It is advisable, especially when brittle steels are used, to pay careful attention to stiffness both in the press bed and in the die holder, even backing up the die steels with other hardened plates to prevent deflection which might permit cracking.

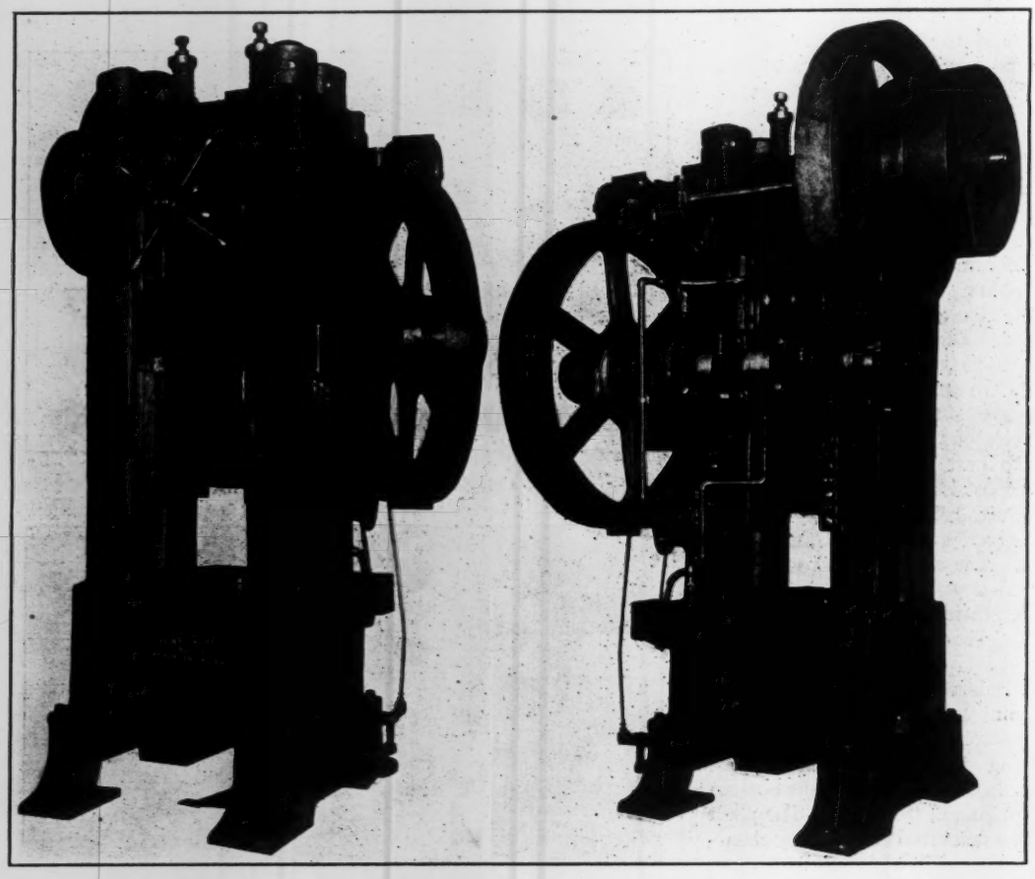

FIG. 16 400-TON KNUCKLE-JOINT PRESS

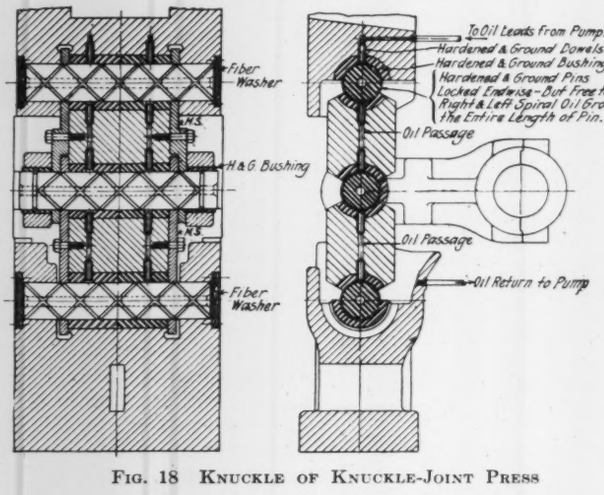

FIG. 18 KNUCKLE OF KNUCKLE-JOINT PRESS

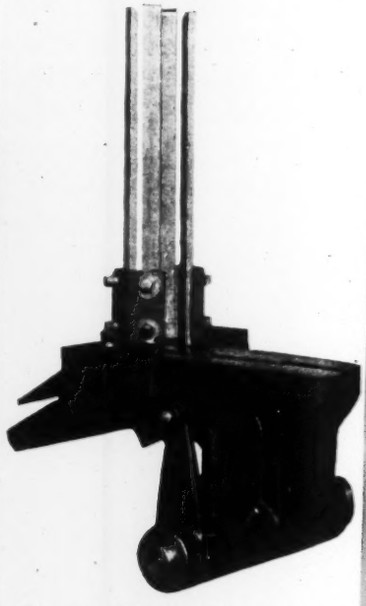

FIG. 19 MAGAZINE FEED FOR PRESS

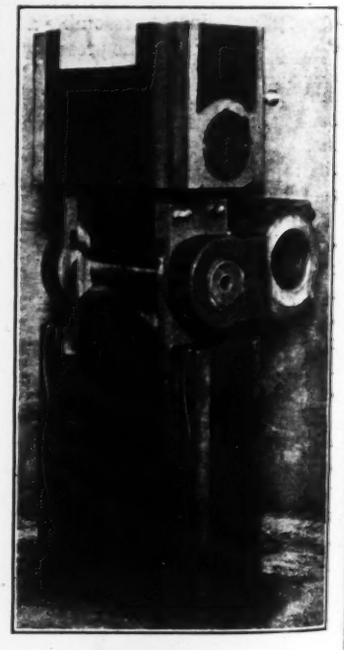

FIG. 17 KNUCKLE OF KNUCKLE-JOINT PRESS

KNUCKLE-JOINT PRESSES

All of the cold swaging and coining work is characterized by a rising load reaching a maximum at the very end of the stroke. This load is effective through a very short distance and since the area of the work is small in proportion to the load, it admits of a proportionately more compact machine than any other type of press work. The modern knuckle-joint press fits into these requirements in every respect. Fig. 16 shows front and back views of one of the latest standard knuckle-joint presses, a 400-ton size, incorporating a number of changes in design, especially of knuckles, gibs, and lubricating system. With rugged frame members tied together with unusually heavy steel tie-rods shrunk-in under a tension equal to the rated capacity of the machine so that there will be no stretch in them up to that point, and with a working mechanism designed to give a large mechanical advantage to the drive, the characteristic press delivers an intense pressure just at the last portion of the stroke where it is needed. By way of comparison, a knuckle-joint press weighing 30,000 lb. may safely be loaded to 600 tons, whereas a 30,000-lb. single-crank press, costing nearly as much, has a maximum capacity of under 200 tons, less than a third. To be sure, the crank press, larger but lighter in

most of its details, has decided advantages for its own class of work.

The knuckles which give the press its name are shown in section in Fig. 17, and in the photograph, Fig. 18. These are bent to raise the slide and straightened to apply the working load by the action of a crank at back of the machine. The bearing pressure through the knuckles is necessarily very much higher than on any other type of press and accordingly the pins and the replacable knuckle bushings are of hardened tool steel, ground and grooved to suit a high-flow, circulating lubrication system which keeps the bearing surfaces bathed in oil. In the recent redesign adapting the old soft-metal coining presses to the more severe conditions of swaging steel, the knuckle pins have been materially increased in size and the area of the knuckle bearing surfaces has been practically doubled. The

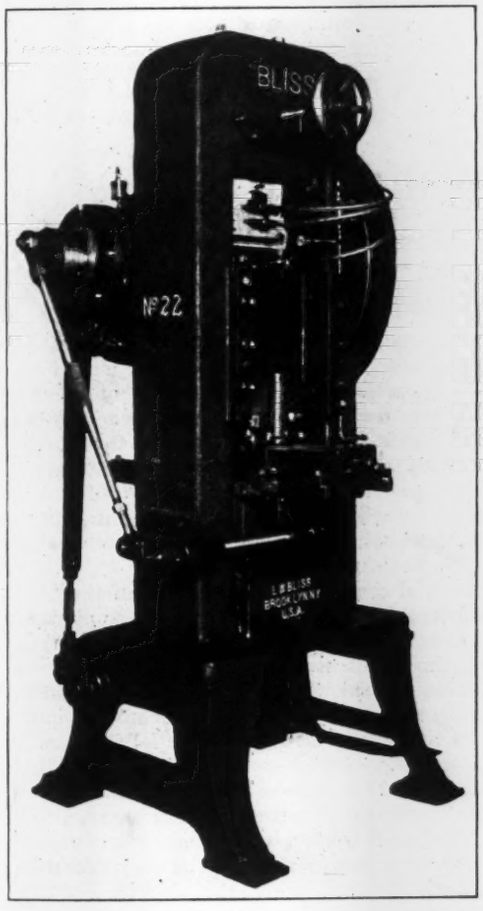

FIG. 20 SOLID FRAME PRESS EQUIPPED WITH COIN FEED

result is a very material increase in the emergency overload capacity of the machine.

In the selection of a knuckle-joint press it is impossible to overemphasize the need for selecting a press heavy enough. The life of the dies, the accuracy of the product, the life of the machine itself depend upon it. It has been well said that a power press is the most abused machine tool there is. Remember that a static load registered on a testing machine is not the same as the load the press gets. The slide is driving down at the rate of thirty, forty, fifty strokes a minute with all the inertia of the flywheel back of it. And if the press is properly put together, the die surfaces will come to the same place every time, practically regardless of what is between them. That is essential if the product is to be held to close tolerances. Further, the blanks, forgings, slugs, strips, stampings, and castings do not come uniform; some are thicker than others, some are harder than others, and beside this, press operators are no more careful than the general run of humanity. One general rule much followed is always to *double the static test load to determine the press capacity required*. Examination of reports of one of the large automobile manufacturers, a pioneer in the sizing of drop forgings, shows that the press selections are made on a basis of about *200 tons per sq. in.* of area to be squeezed. Another automobile concern, starting out with 250-ton press which did the work, jumped to a 400- and then a 600-ton machine getting increased die life and accuracy each time.

AUTOMATIC FEEDS

Sizing operations are fast as a rule, because the pieces do not have to be accurately placed on the die. Good hand-feeding practice appears to range between ten and thirty pieces per minute. For the swaging and coining operations the work usually has to be accurately placed and the rate is lower, say five to fifteen a minute. On almost any of this work, however, where the production is large enough to warrant it, the application of a mechanical feed will materially increase the output. Table 2 is a reproduction of a

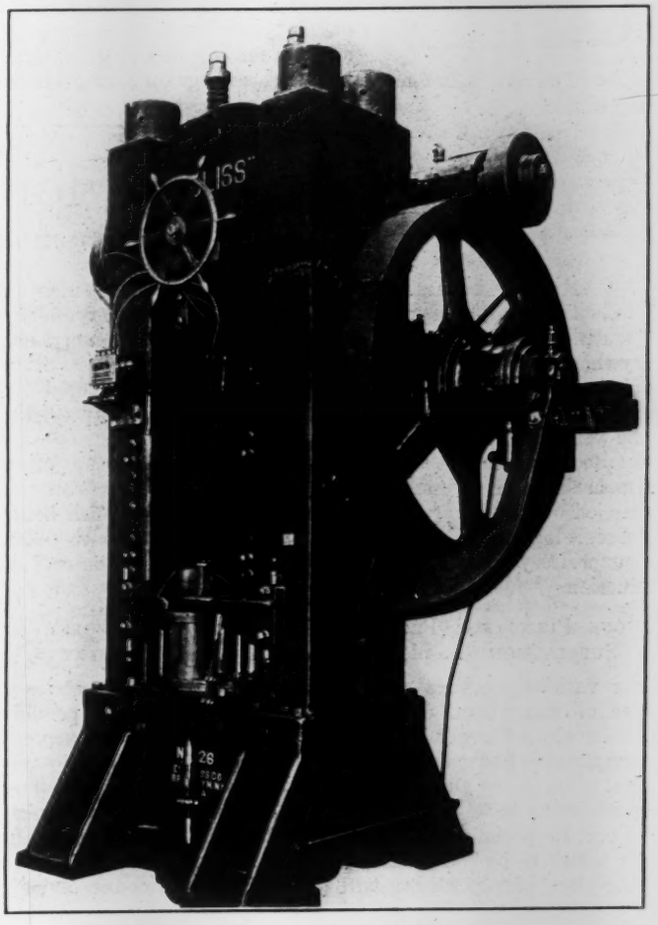

FIG. 21 PRESS EQUIPPED WITH RACHET DIAL FEED

table form Mr. Kelso's article of 1924 showing increases recorded at the Hudson Co.

TABLE 2 COMPARISONS OF INCREASED PRODUCTION
(From A. R. Kelso)

Group number	Part name	Percentage of Increase Over the Machining Method	
		Manual feed	Mechanical feed
1	Accelerator lever	387	583
4	Accelerator lever	430	537
5	Brake lever	1042	1864
11	Brake lever	780	944
28	Clutch yoke	358	894
32	Fan supporting yoke	624	774

The advantage of the mechanical feeds which are adaptable to this work, are that they place the work accurately under the slide, catching every stroke of the continuously operating press, relieving the operator of considerable fatigue and of certain nervous strain from the danger of getting his fingers caught. Those mechanical feeds which may be used to advantage in different cases include the simple magazine feeds, known also as coin feeds or push feeds, and some station dial feeds, chain feeds, and friction dial feeds. Occasionally hopper feeds also are used but these, as a rule, are very special development jobs.

Fig. 19 shows a simple type magazine feed ready for attachment

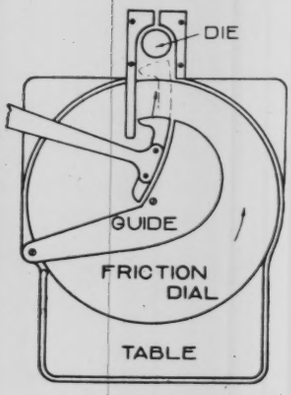

FIG. 22 FRICTION DIAL FEED

to the press. This consists of a receiver for a stack of blanks and a reciprocating slide to push one blank off the bottom of the stack into the die each stroke of the press. Fig. 20 shows a solid frame, old type press equipped with a regular coin feed for Mint service. The principles of this feed are similar to those of the magazine feed. Fig. 6 shows front and rear views of a recent 250-ton press equipped with a reciprocating slide feed without the magazine. Various modifications of this type of feed are used especially for feeding and ejecting forgings and castings for sizing jobs.

The station or ratchet dial feeds are especially suitable for accurate positioning of irregular articles. Fig. 11 illustrates one of these feeds on another type of press where two odd-shaped pieces are assembled by a swaging-riveting operation previously described. Fig. 21 shows an 800-ton press fitted with a dial feed and used for upsetting brass shell cases.

Fig. 22 shows diagrammatically the principles of the friction dial feed. This type is adapted especially to simple round blanks or cups such as button blanks. These are straightened by hand on the table at the front, lined up and kept moving in by the friction disk; and fed to the die by one of two or three different motions. The gages at the die must also be adapted to the job.

Chain feeds are of many types, all of which are usually comparatively inexpensive, designed on the conveyor principle for work which does not require great accuracy in positioning. They are driven like other automatic feeds from the press crankshaft.

Training for Foremanship

BY FRANK CUSHMAN,[1] WASHINGTON, D. C.

IT CAN fairly be assumed that the supervision of men in industrial employment is fundamentally necessary. The truth of this proposition is supported by industrial experience everywhere in all types and classes of manufacturing activities. No industrial organization has ever successfully conducted its business without making some definite provision for supervision of the workers employed.

Granted, therefore, that the supervision of men on the job is indispensable and necessary, it follows that this supervision can be carried out with varying degrees of effectiveness. Such being the case, it logically follows that the degree of efficiency with which the supervisory function is performed directly affects the cost of production.

VARIOUS PLANS FOR PROVIDING THE NECESSARY DEGREE OF SUPERVISION OF MEN IN INDUSTRIAL EMPLOYMENT

The various methods and plans for providing the necessary degree of supervision exhibit almost every variety of possible plans in almost every conceivable combination. For purposes of comparison, however, they may be roughly classified into two general groups, as follows: (1) Those in which the controlling purpose seems to be to subdivide the supervisory job and assign parts of it to special agencies; and (2) those in which the controlling factor seems to be the development of a high degree of efficiency on the part of supervisors with the minimum amount of subdivision of supervisory responsibilities.

Those plans which fall into the first group ultimately result in a highly functionalized organization. Such a development means an increasing number of non-productive workers, increased overhead, and, if carried to an extreme stage, a very minute subdivision of the supervisory job. To an increasing degree good personnel relationships are absolutely necessary as the supervisory job is subdivided, and a highly functionalized organization necessitates a high degree of coöperation between all of the individual agencies or offices, which must work together if efficient supervision is to be secured.

The extreme case of the second type is nothing more than the old-fashioned organization where a supervisor or foreman has all of the responsibilities that belong to the supervisory job. In extreme cases which are still to be found in many small organizations the foreman is his own employment manager, hires and discharges men as needed, plans his own work, and carries all the detailed responsibilities that go with the immediate supervision and management of the work. This may include even the instruction and training of apprentices and green men.

[1] Chief, Industrial Education Service, Federal Board for Vocational Education.

Contributed by the Committee on Education and Training for the Industries and presented at the Providence Meeting of the A.S.M.E., Providence, R. I., May 3, 1926.

CYCLES THROUGH WHICH INDUSTRIAL ORGANIZATIONS PASS AS REGARDS FUNCTIONAL ACTIVITIES

In actual practice we seldom, if ever, find examples of either extreme type. Industrial organizations, no matter how highly functionalized they may be, find it absolutely necessary to use foremen as supervisors. These foremen are found to be indispensable notwithstanding the fact that in the development of industry, auxiliary agencies for supervising, planning, and managing the work of production have been introduced into the organization. On the other hand, we seldom find an organization where the foreman carries the entire list of responsibilities that belong to the job of supervising men.

As an industrial organization grows, one of the first auxiliary or functional departments to be added is the employment department. This is accompanied or followed by a purchasing department with a purchasing agent who is made responsible for buying stock, supplies, and equipment, and in many cases is held responsible for the quantity and quality of all materials brought into the plant. Such agencies in an organization may be followed by a planning department, where a staff of specialists takes over the responsibility of planning work in more or less detail in routing jobs through the shops, an inspection department which assumes the responsibility for the quality of work, a tool engineer responsible for keeping tools and fixtures in proper shape, a training department which takes over the responsibility for breaking in new workers and training apprentices, and so on through the entire list of auxiliary agencies which may be added to the organization from time to time.

Observation of industrial organizations throughout the country has confirmed the general opinion that industrial organizations pass through a series of cycles as far as functional activities are concerned. A given organization will start business under a very simple plan where competent foremen are held responsible for supervising the work of their departments. As the organization grows, auxiliary activities will be added little by little until a point is reached where the overhead expense becomes too great. When this point is reached the organization usually passes through a stage that may be termed the "house cleaning" period. At this point most or all of the functional activities are thrown out, the number of persons performing supervisory work is reduced to the absolute minimum, and foremen are secured who can carry the responsibilities that go with the foreman's job. In many cases, after such a reorganization and simplification in the management, additional steps will be taken to functionalize the organization once more.

The determining factor back of all of these changes in industrial organizations is necessarily the cost of production, and ultimately the plan must be followed which results in the greatest efficiency and economy. Plans of either of the two general types previously

discussed will be followed, modified, or abandoned in proportion as the management believes that the cost of production will be reduced thereby.

THE FOREMAN AN INDISPENSABLE FACTOR IN ANY INDUSTRIAL ORGANIZATION

At the present time it is being recognized to an increasing degree that the foreman is an indispensable factor in any industrial organization. Human nature and human motives have not changed and are not likely to change very much in spite of the fact that science and invention have changed working conditions and industrial processes to such a degree that the "iron man" seems to be the principal human or inhuman factor in industry today. Regardless of the degree to which an industry is mechanized, men are still necessary, and industrial executives are, to an increasing degree, recognizing that no scientific or pseudo-scientific plan of personnel management can wholly displace the foreman. He is the man who stands between the management and the men on the job. The improvement of foremanship is therefore one of the problems in which there is a nation-wide interest at this time, and the question as to what can be done to help the foreman in industry is of major importance.

In considering any plan for helping the foreman the first logical step would seem to be to make a rough inventory of his assets and liabilities. Among his assets the following may be mentioned. He knows the details of the job better than any one else can know them. He is accustomed to getting results and uses his practical judgment and skill, resulting from long experience on the job, in a way that a person who has not had a large amount of first-hand experience on practical jobs cannot fully appreciate. He knows his men as a rule better than any one else in the organization, and he deals with them on the job with regard to their individual characteristics and such qualities and abilities as they may have. If he continues as a foreman it can be assumed that he has at least the minimum knowledge and skill necessary to be an acceptable foreman. This is true whether he has anything more than a common-school education or not, so far as general education is concerned. In taking care of all of the practical details of getting the work out, he utilizes all sorts of practical ideas in overcoming difficulties and dealing with practical situations that the superintendent may know little or nothing about. This statement is not intended in any way to cast a reflection upon the degree to which a superintendent knows what is going on in his plant. The very nature of his relationship to the work is such that it is impossible for him to have a working command of all the practical details of getting jobs done in the same way that the foreman has this command.

The foreman's assets so far overbalance his liabilities that a discussion of such liabilities as he has is in some danger of being misunderstood. Instead of using the word "liabilities," it would perhaps be better to refer to the specific items as points on which experience has shown a foreman can be helped most effectively. Because of his practical training and experience the foreman's knowledge is exceedingly specific. He thinks in terms of concrete experiences rather than in terms of general principles. It is probably true that he lacks ability to organize what he knows in such a way as to utilize his experience most effectively in dealing with special situations. Such a condition is to be expected where men have been held closely in their work for a long period of years and have never had an opportunity to analyze their jobs or responsibilities nor to organize the knowledge which they have regarding their work. Another characteristic of the typical foreman is a tendency to handle situations by rule-of-thumb methods. He has probably developed the habit of making decisions or dealing with situations on the basis of a "hunch" rather than on the basis of a careful consideration of all the elements which affect a particular problem. Observation of large numbers of foremen and the study of their work in industrial organizations furnish a basis for the statement that the foreman is a good deal of an individualist. The average foreman tends to place the individual success of his own particular department or division of the plant above everything else, and is likely to subordinate anything or everything in order to keep his own department going in first-class shape.

THE FOREMAN'S NEEDS IN THE WAY OF EDUCATION OR TRAINING

In view of the preceding statements, the question naturally arises, What does the foreman need in the way of education or training? What form or type of educational activity is likely to be of maximum service? In an attempt to answer partially this question the following items are given as suggestive of what appears to be the most important immediate field for development so far as education and training are concerned: (1) Help the foreman in organizing what he already knows so that his great reservoir of experience and practical knowledge can be utilized for all that it is worth. (2) Help him to secure a better appreciation of his responsibilities (*a*) as a member of a large organization, (*b*) as a leader of men in his own department, (*c*) as a unit in the management, (*d*) as an efficient supervisor of work, and (*e*) as an instructor on the job.

It should be noted that neither general educational subjects nor technical subjects are included in the preceding statement. There will undoubtedly be a wide divergence of opinion relative to what foremen need the most in the way of education and training. This paper is based upon personal experience, observation, and first-hand educational work with a large number of practical foremen. It is submitted, therefore, not as the last word on the subject, but as a point of view based upon experience and certain conclusions which have been arrived at as a result of that experience.

The principal educational needs of foremen, previously mentioned, make it possible to set up rather definite eduational objectives for foremanship courses. The careful and thorough determination of the objectives back of any form of education and training for foremanship is of course the first step. The second step is to consider ways and means for attaining these objectives. Generally speaking, there are three principal types of educational procedure. First, we have the procedure of imparting information. Information may be imparted through lectures or through distribution of books, papers, or pamphlets. Imparting of information by any of the possible agencies is a very important and generally approved method of carrying on certain types of educational work in schools, colleges, and industry itself. Certain educational objectives can be adequately handled by supplying accurate and complete information. Other educational objectives cannot be handled at all through this procedure. The imparting of information or any of the methods of instruction of use in carrying out the procedure have little or no value in connection with handling any of the foremanship objectives indicated in the preceding discussion.

A second educational procedure consists of organized instruction through the use of recognized and accepted teaching methods. This procedure is suitable where the objective is to bring a learner up to a point where he can do something which he was unable to do before being taught. The instructional procedure is in turn well suited to the attainment of certain educational objectives. For others it is not at all suitable.

THE CONFERENCE AS AN EDUCATIONAL PROCEDURE

The third type of educational procedure is now commonly referred to as the "conference," although it closely resembles the problem method of teaching. The conference as an educational procedure consists essentially of six steps or stages, as follows:

1 The assembling of facts
   a From first-hand experience (our own experience)
   b From second-hand experience (other people's experience)
2 The selection of functioning facts
3 The evaluation of functioning facts
4 The making of some sort of a decision or arriving at a conclusion
5 The making of a plan to carry out the decision
6 The carrying out of the plan.

The conference as an educational procedure consists essentially of a systematic thinking-through of problems. The first step is the assembling of facts that have some bearing on or some relationship to the problem under discussion. The second step consists of selecting from the assembled facts those particular facts

which have a direct bearing upon the problem. The third step in the conference procedure is the evaluation of these functioning facts. As a result of this evaluation, members of the group evolve ideas relative to the problem. These ideas are modified as a result of the ideas which are expressed by other members of the conference.

This process of evaluation may go on for a considerable time until members of the group reach some sort of a decision in regard to the problem before the conference. The decision with respect to any given problem may or may not be unanimous as far as the members of the conference are concerned. It is not of particular importance in a foremen's conference that the decision should be unanimous on all questions. It is more important that each and every member of the group shall have done some thinking relative to the problem than it is that all members should arrive at the same conclusion. Any well-conducted foremen's conference should carry the conference procedure through the first four stages or steps as given in the preceding outline.

Under certain conditions the making of a plan for the purpose of carrying out the decision is handled in a foremen's conference. In other cases this phase, which is the fifth step of the conference procedure, may be handled subsequent to the conference work with a group of foremen. The sixth step of the conference procedure may extend over a period of months or even years, and the actual results of a conference in a given plant may be more apparent several years after the original foremen's conference was held than they were immediately following the completion of a series of foremen's conferences in the organization.

The idea has been accepted in certain cases that a conference was a sort of unsystematic "gab-fest" and that the work was carried on without any particular plan and amounted to little more than getting a group of men together for the purpose of carrying on a more or less aimless discussion with respect to phases of their jobs. Such a conception is absolutely incorrect where a conference is carried on under competent leadership. In a well-conducted conference the essential feature of the work is the carrying out of the principles of constructive thinking relative to specific problems. The leader of a foremen's conference does not teach subject-matter in the sense that an ordinary school teacher teaches it. The subject-matter of a conference is drawn from the experience of the members of the group, and the ideas or decisions arrived at by the conference group are modified in particular instances by the collective experiences of all of the members of the conference. To sum up, it may be stated that the conference as an educational procedure is primarily directed toward helping the group members to organize the knowledge which they already have concerning their jobs, and to help them to acquire the habit of intelligent constructive thinking relative to their jobs and their responsibilities.

Soundness of Theory Back of Conference Plan of Foreman Training

Experimentation with the conference plan of foreman training in a great variety of industries in all sections of the United States has indicated very clearly that the general theory back of the procedure is sound. Regardless of what additional education or training foremen may need, it has been demonstrated that the conference is one of the best possible points for beginning an educational program. A foremanship conference, if well handled, places the foremen in such an attitude of mind that they are keenly interested in other educational activities in which it might be difficult or impossible to interest them if the conference were not used as a preliminary procedure. Regardless of what other forms of instruction may have value for foremen, the conference procedure has a definite value and should be retained permanently as a phase of whatever program of education and training may be set up for foremen in any industrial organization.

From a theoretical standpoint it would seem to be possible to prepare young men for foremanship through the proper types of preëmployment training. Practically we find industry under the supervision of practical men who have grown up in industry and won a place for themselves as foremen as a result of the possession of certain qualities of leadership and a thorough knowledge of their work. There seem to be no grounds for assuming that this situation is going to be changed materially in the immediate future. Prospective foremen are constantly being groomed for their jobs in the hard school of experience, and it is from the ranks of skilled and competent workers that management is securing its foremen. Such being the case, it would appear to be more important to take the men who are on the job as foremen and to do the best possible job, through wisely directed forms of education and training, to prepare them for increased efficiency on the jobs which they have, than to ignore this tremendously important group and concentrate attention upon prospective foremen who are still training in school or college.

In other words, the most important phase of the job of training for foremanship is, at present, the development of suitable trade-extension opportunities for employed foremen. The foremen now on the job constitute one of industry's most important and valuable assets. A program of education and training which will help these men to bring themselves up to a point where their abilities are utilized to a maximum degree is one of the great industrial economies which is only beginning to be appreciated.

The Santa Fé Apprenticeship Plan

THE Santa Fé apprenticeship plan was inaugurated in September, 1907, and has been in operation over 18 years. It was distinctly the outgrowth of a situation in which the carrier could not secure sufficient skilled mechanics at any price, and consequently undertook to train them. Of this plan the *Railway Review* says in a recent editorial: "If all the railroads in the country would maintain as efficient a training system as the Atchison, Topeka and Santa Fé Railway, the main problem on our railroads would soon be solved."

The object of the apprenticeship course is primarily to develop fully qualified mechanics in various lines of work and secondarily to establish a more permanent working force. At present there is an apprentice school at practically every shop or round-house point, instruction being given in nine trades. On July 31, 1923, the total number of unskilled apprentices was 1982.

There were three types of apprentices, (1) regular apprentices, i.e., boys over 16 years of age, (2) helper apprentices, i.e., boys chosen from among those who have served one year as a helper, (3) special apprentices, i.e., graduates from a technical college. The first of these groups is endentured for four years, the other two for three years each. Applicants must, in addition, pass a medical examination and a simple mental test. The majority of the boys come from the community in which the shops are located or from small outlying towns.

Both practical (shops) and theoretical (classroom) training is given. During the period of his training an apprentice is expected to cover the processes of an entire craft, a specific amount of time being assigned to each operation. Shop work is under the supervision of some 80 shop instructors, each of whom has charge of about 25 boys. All apprentices other than special apprentices are required to spend 204 hours per year in the classroom, that is, two periods of two hours each week after working hours. The chief subjects taught are mechanical drawing and shop mathematics.

Most of the instructors are graduates of the apprentice school. At the large points regular meetings of instructors are held to discuss the progress of apprentices. A meeting of all shop instructors on the system is held annually.

A progress record card is kept in duplicate for each apprentice, one of which is filed in Topeka and the other at the local shop point. These indicate the amount, kind, and quality of work that is done. This also provides a check against keeping a boy too long upon one kind of work.

In each shop there is an Apprentice Board composed of the general foremen, the department and gang foremen, the shop instructors and the school instructor. Each apprentice is brought before this board either in person or by name at intervals of six months throughout his training. Recommendations are then made to the ranking mechanical officer of the shop regarding matters of work, discipline, promotion, transfer, etc. These Boards are said to have had very fruitful results in enlisting the interest of shop officials in the personal welfare and progress of each boy. (Industrial Management, vol. 72, no. 1, July, 1926, pp. 22–23.)

Sane Specifications and Intelligent Inspection

Problem of Tolerances to Be Allowed in Projectile Manufacture Can Be Solved only by Experimental Study of Effects of Dimensional Variations

By BRIG.-GEN. C. L'H. RUGGLES,[1] WASHINGTON, D. C.

THERE are so many different kinds of ordnance material, each having its special problems, that I find it desirable in discussing this subject to limit myself rather largely to a single class, although some of my remarks will also apply equally well to ordnance material in general. I have selected mobile-artillery ammunition as the basis for my discussion on account of its requirements for large mass production and the special considerations required for satisfactory functioning. I believe that this limitation which I have placed on my subject will appeal to you because mobile-artillery ammunition is the largest item to be manufactured in any emergency, and manufacturers will become more acquainted with this type of material than any other. Many of you during the World War probably were engaged in the manufacture of some of the metal components that make up a complete round of ammunition.

Difficulties Arising During the Wolrd War

During the World War the Ordnance Department was severely criticized because of the requirements of its drawings and specifications, and also because it was felt that inspection was not being carried onintelligently. Much of this criticism was justified, but some of it was unjust because it was made without a full realization of the use and requirements for the articles in question. It is my purpose to summarize briefly some of the principal difficulties during the World War and to point out the steps by which we hope to insure "sane specifications and intelligent inspection."

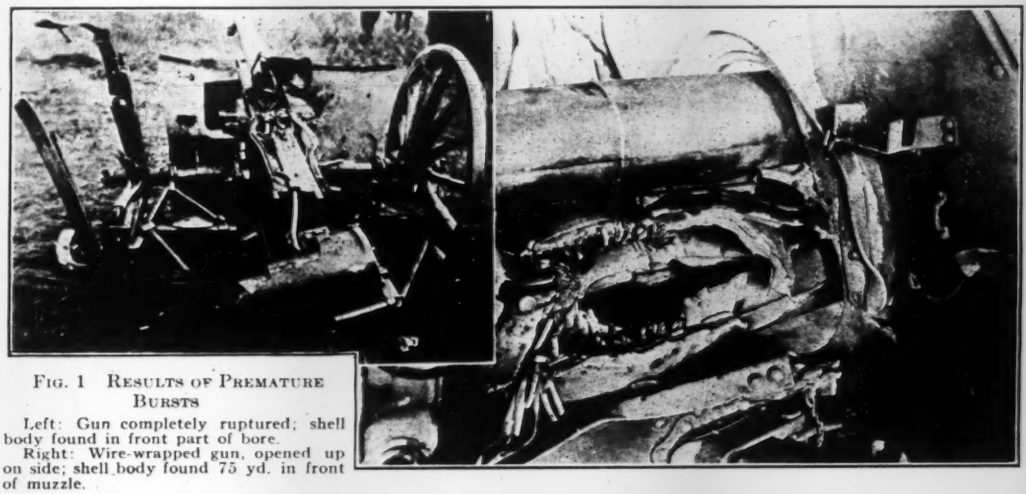

Fig. 1 Results of Premature Bursts

Left: Gun completely ruptured; shell body found in front part of bore.
Right: Wire-wrapped gun, opened up on side; shell body found 75 yd. in front of muzzle.

Sane specifications and intelligent inspection require full knowledge as to the essentials necessary to a good usuable product as against refinements which might be desirable but which add too little to the serviceability of the product to justify the added difficulty of production. Such refinements are frequently added to cover doubtful points or points on which there is no information, and are therefore factors of safety, frequently referred to as factors of ignorance. The only satisfactory method of eliminating these refinements is to demonstrate by actual tests that they are not necessary. This is simply saying that the article should be completely developed before an attempt is made to produce it in quantity.

Failure to have completely developed types of our own at the outbreak of the war, thus requiring development and production to be carried on simultaneously or the adoption of foreign types concerning which we had little or no information, was the cause of many difficulties encountered in our wartime production programs.

It is difficult to obtain definite information on the essential requirements of ordnance material due to its infrequent use under actual service conditions and because when actually used it is under conditions that make the recording of information, as to its functioning, difficult if not impracticable. Tests are of course made at a proving ground, but large-scale tests are expensive, so that it is generally necessary to adapt the test programs to the funds available rather than to the technical requirements. We do not have in time of peace the same supplementing of shop and proving-ground tests by actual service use as is obtained with most commercial products. Therefore the same degree of development, which would largely eliminate the necessity for changes either to correct defects in functioning or to facilitate manufacture, is not practicable. There is of course some use of equipment by troops in training that gives valuable information, but such use is necessarily limited and cannot closely conform to battle conditions.

Prewar Designs Largely Developed Through Small Yearly Production Programs

Our prewar designs were largely developed through small yearly production programs and no great difficulty was encountered in producing to the small tolerances prescribed. Even orders placed with private manufacturers were in such quantities as not to justify mass-production methods. No demands for larger tolerances arose under such conditions; therefore, when the large-scale production, which did call for larger tolerances and concessions on quality of material, started, it found the Ordnance Department without the necessary data upon which to base intelligent decisions. The use of artillery and the corresponding expenditure of ammunition during the war was on a scale not previously anticipated. This required frequent decisions based entirely on snap judgment, and accounts for many of the peculiar experiences of manufacturers; where one decision might be given at one desk, it might be reversed at the adjoining desk in the same office, and perhaps there might be several reversals before the final decision of the chief of the division or the Chief of Ordnance was given.

Another condition which was responsible for many of the peculiar decisions was the lack of experience of the officers making the decisions. Due to the lack of a trained reserve, it was necessary to place in positions of responsibility temporary officers with little conception of the requirements of the material for which they were responsible. Even in cases where officers had a certain amount of experience, it was usually with some particular component. The broad background necessary in order to know what effect changes made in a particular component would have on the functioning of the completed article or on the related equipment was almost entirely lacking. Due to the same condition of lack of previous training there were of course many assignments of personnel to positions which they were entirely unfitted by temperament to fill with success.

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CONFUSION DUE TO ADOPTION OF FOREIGN DESIGNS AND SPECIFICATIONS

It might be assumed that, with American manufacturers having two or three years' experience on manufacturing material for the Allies, and our subsequent adoption of many of those foreign designs, most of the difficulties would have been already solved. The experience of American manufacturers on those foreign orders was undoubtedly of incalculable assistance to the Ordnance Department, just as was the adoption of foreign designs and specifications. The confusion arose from the necessity for adopting the designs of so many countries, no two of which agreed on some of the basic principles. All bore evidences of yielding in rigid requirements to meet the conditions of quantity production, but the concessions were on different features and were entirely lacking in consistency. There were designs of articles of the same general class, particularly artillery ammunition, some distinctly American, some British, some French, some Russian, and frequently some showing the influence of German designs, and there was considerable lack of consistency as regarded the tolerances permitted on the various dimensions of projectiles between the different designs.

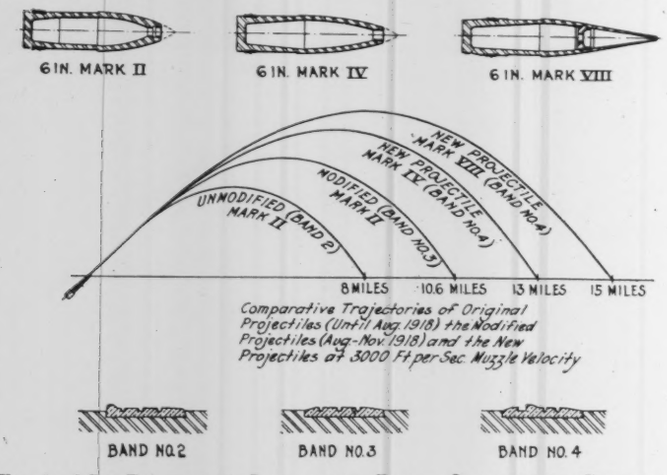

6 IN. MARK II
6 IN. MARK IV
6 IN. MARK VIII
NEW PROJECTILE MARK VIII (BAND NO.4)
NEW PROJECTILE MARK IV (BAND NO.4)
MODIFIED (BAND NO.3) MARK II
UNMODIFIED (BAND 2) MARK II
8 MILES
10.6 MILES
13 MILES
15 MILES
Comparative Trajectories of Original Projectiles (Until Aug. 1918) the Modified Projectiles (Aug.-Nov. 1918) and the New Projectiles at 3000 Ft per Sec. Muzzle Velocity
BAND NO.2
BAND NO.3
BAND NO. 4

FIG. 2 6-IN. PROJECTILES INCREASE IN RANGE OBTAINED BY CHANGING BAND DESIGN

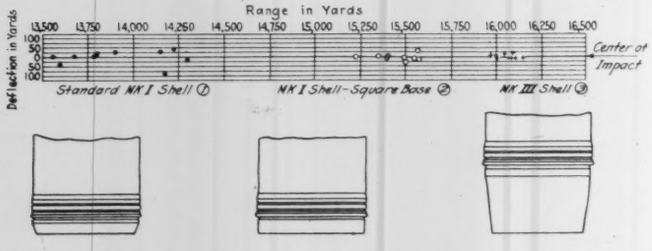


FIG. 3 240-MM. PROJECTILES EFFECT OF DIFFERENT FORMS OF BASES ON RANGE AND ACCURACY
(Muzzle velocity, 1700 ft. per sec. approx.)

The Ordnance Department designs generally showed a close fit between the bourrelet diameter and the bore of the gun with rather small tolerances. Our designs also had a rather close tolerance on the rotating-band diameter and on its location with reference to the base. The British design, on the other hand, generally had no bourrelet, depending upon the body diameter to center the projectile in the bore. Both the clearance and the tolerance were comparatively liberal. The British also permitted a liberal variation in base thickness and in location of the rotating band. They furthermore permitted a method of repairing light-weight shell by adding a metal plate to the base, which still further changed the location of the rotating band and the thickness of the base. The British were, however, particular as to the dimensions of the rotating band proper and the finish of the projectile, especially that of the interior, which came in contact with the explosive. The French were intermediate between the British and the American on bourrelet tolerance, somewhat closer than either on the rotating band diameter, and quite generally particular on the uniformity of location of the center of gravity. The Russians at the early stages insisted on a very high grade of machine finish, even requiring nickel plating. The result of this evident lack of a consistent and well-established system of tolerances was that there was a tendency on the part of the ordnance designer, who considered the matter largely from the quality standpoint, to lay down types, using as tolerances on each dimension the smallest which he could find on any of the various designs, American, British, French, or Russian. The manufacturers, on the other hand, interested largely in quantity output, could not see why if a large tolerance was permitted on a certain dimension on a British design, an equally large one should not be satisfactory on the other types. Similarly, if large tolerances on other dimensions were satisfactory on the French design, they should be equally satisfactory on other designs—all of which resulted in a design carrying on each particular dimension the largest tolerance that could be found on the corresponding dimension of any of the foreign designs. Those who were familiar with the French designs that we adopted, such as the 75-mm., 155-mm., and 240-mm., will recall that these shell had relatively thin walls and large bursting charges while the British 8-in. and 9.2-in. designs had much thicker walls and smaller bursting charges. With types so markedly different, and with lack of information on the proper tolerances, it was obviously impracticable to adopt a composite design with the tolerance on each dimension the largest that could be found on any particular dimension of the British, French, American, and Russian designs. It was assumed that these French and British designs which were being used before we entered the war must have been carefully developed and that no great departure in the tolerances could be permitted.

PRESENT PROBLEM OF ORDNANCE DEPARTMENT IN REGARD TO TOLERANCES

It is the present problem of the Ordnance Department to work out designs and specifications which are proper compromises between the quality and quantity viewpoints, and in this work the assistance of practical manufacturers is needed. As fast as it can be shown that tolerances on certain of the dimensions can be increased without appreciable loss of effectiveness, then such increases will be made. On the other hand, when it is shown that small tolerances are essential, it is believed that manufacturers will proceed to work out methods for obtaining these refinements as they did during the World War, when the necessity could actually be demonstrated.

REQUIREMENTS TO BE MET IN DESIGNING ARTILLERY AMMUNITION

In designing artillery ammunition every effort is made to meet the very definite requirements which the artillery service desires. A projectile must first of all be capable of being loaded into the gun and subsequently pass through the bore. This is the simplest requirement and demands only careful gaging of each projectile. Another requirement of prime importance is safety, that is, the projective must not have defects which might cause it to burst prematurely in the bore. Too much insistance cannot be placed on this requirement, and the standard must be higher than on almost any other product. In ordinary commercial products a reasonable percentage of failures is permitted, since the consequences are ordinarily not particularly disastrous, and only result in loss of time while repairs are being made. A premature burst, however, usually results in the complete destruction of an expensive piece of artillery, and frequently in the death of the gun crew as well. (Fig. 1.) Moreover, the resulting loss of confidence in the ammunition, affects the entire morale of the army.

On account of the disastrous results of a premature burst, a high degree of excellence is essential on all points affecting safety. Premature bursts may be due to several causes, such as defective shell, defective fuse, defective explosive in the shell, or defective loading of the explosive. With premature bursts possible from any one of these causes, as many as one such burst in twenty thousand rounds fired is considered only fair performance. The British had periods where there was less than one for two hundred thousand rounds. There also are records which indicate that during the early months of the war one of the allied countries had as many as forty premature bursts in one day, and at times it was necessary to gage the bourrelet of every shell before it was loaded in the gun.

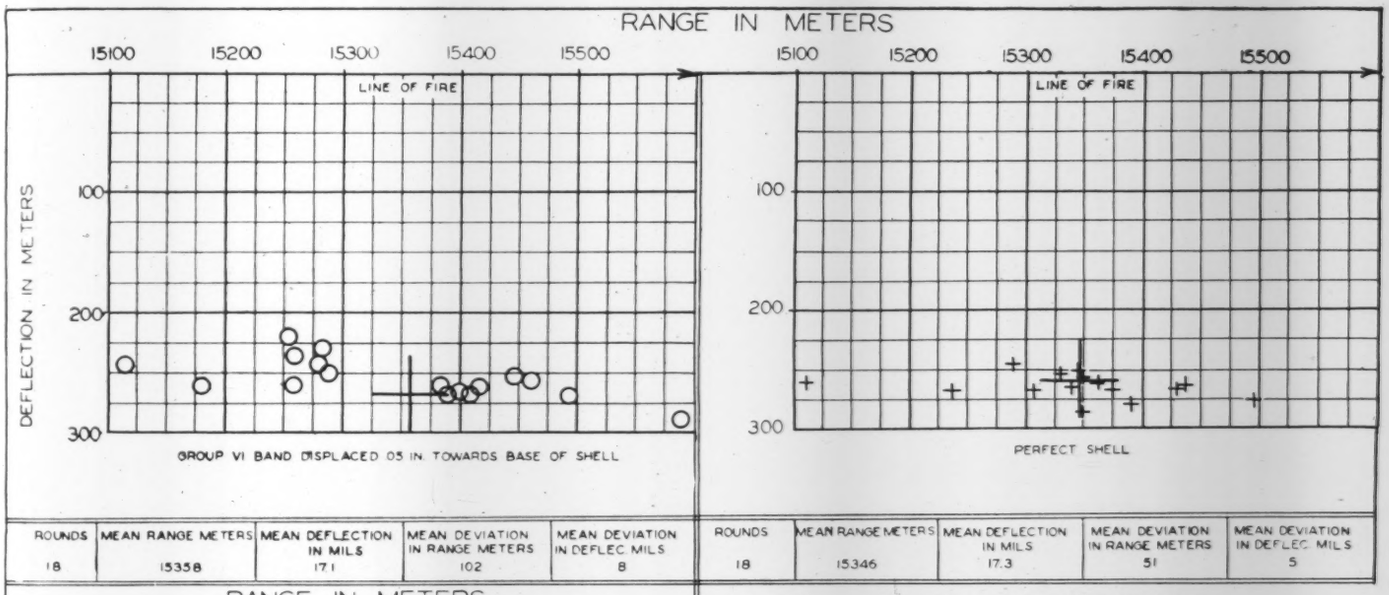
EFFECT OF SMALL CHANGES IN PROJECTILE DESIGN

Uniformity of ranging is another essential requirement, and this is perhaps the one that more than any other controls the tolerances and also is the one concerning which the greatest uncertainty exists. A projectile passing through the air at the rate of something like half a mile a second or eighteen hundred miles an hour meets a high air resistance, and a number of projectiles thus traveling through the air for a distance of from ten to thirty miles must have uniform resistance in order that they may range alike. This means uniform dimensions, finish, balance, etc. Very small changes in contours may have surprisingly large effects on range and accuracy. The effect of a small change in the shape of the rotating band has been the subject of frequent publication as illustration of this fact, but will bear repetition.

Referring to Fig. 2, it will be seen that by changing the band design from Type No. 2 to Type No. 3 an increase in range was obtained of 2.6 miles, and that later by change in the shape of the projectile still further increases in range were obtained.

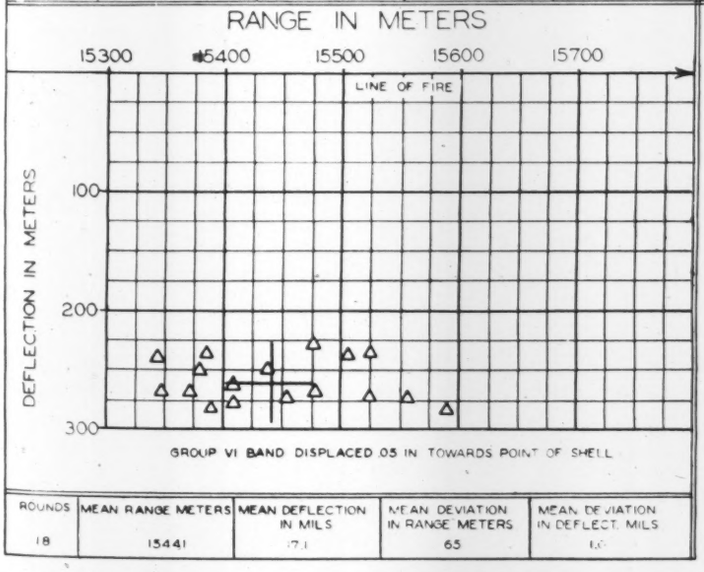
The effect of a slight change in the contour at the base end is illustrated by a change in design authorized to assist manufacturers and permit the use of defective forgings that did not fill out on the corner near the base. No doubt many can recall decisions during the late war based upon snap judgment where reasonable concessions were not approved, yet here is a case where an apparently harmless concession was granted, with the result that the range and accuracy were so adversely affected as to make the projectiles practically useless.

Fig. 3 shows at the left the type of base authorized to assist the manufacturers in production, and directly above it the range and dispersion of a group of these projectiles. In the center is shown the original design, and the range and dispersion of a group of this type. The type at the right is a modification made after the war in an effort to reclaim the projectiles of wartime manufacture which were so unsatisfactory. This modification resulted in a still further increase in range. The most marked feature, however,



ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS	MEAN DEVIATION IN RANGE METERS	MEAN DEVIATION IN DEFLEC. MILS
18	15358	17.1	102	8

ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS	MEAN DEVIATION IN RANGE METERS	MEAN DEVIATION IN DEFLEC. MILS
18	15346	17.3	51	5



ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS	MEAN DEVIATION IN RANGE METERS	MEAN DEVIATION IN DEFLECT. MILS
18	15441	17.1	65	1.6

COMPARATIVE TARGET RECORD

FIRING OF 155 MM. MK III SHELL TO DETERMINE THE EFFECT OF TOLERANCES IN POSITION OF BAND OF SHELL AT 30° 35' ELEVATION AND ABOUT 2410 F.S. VELOCITY ON SEPT. 3, 1920 UNDER O.P. 4084 A.P.G. SEPT. 8, 1920 CLASS E DIV. 3 NO. 155 G – 8

FIG. 4 EFFECT OF DISPLACEMENT OF BAND TO THE FRONT OR REAR

is the decrease in range of some 2000 yards and increased dispersion which occurred when this seemingly harmless concession, which permitted a 20-deg. angle at the base, was granted to assist the contractors in using forgings which would not clean up at the base. The same concession was granted on other calibers of shell such as the 4.7-in., the 6-in. and the 8-in. with the same result.

After a few experiences such as this, it can readily be seen why there was so much hesitancy in granting seemingly simple concessions that would have permitted the acceptance of material which would otherwise be rejected.

It is of course idle to presume that the tolerances given by the drawings or specifications represent the absolute limit for a good usable product. In fact, in practically no case is there a definite dividing line between a satisfactory and an unsatisfactory product of this kind. There is usually a gradual falling off in the accuracy or other requirements, making it necessary to fix more or less arbitrary limits.

In order to prepare sane specifications and practical designs, and in order to more nearly determine the practicable limits for projectile dimensions, a large number of 155-mm. projectiles have been made and tested which had certain important dimensions varied beyond the present limits. While the results of these tests are not conclusive, they indicate the importance of holding certain dimensions as close as practicable. The particular dimensions fixed by the drawings were not necessarily better than other dimensions which might have been stipulated, but great variation was generally not permissible. For example, the rotating band of a projectile might be moved forward or back a considerable distance without seriously affecting the accuracy of a group of projectiles, all having

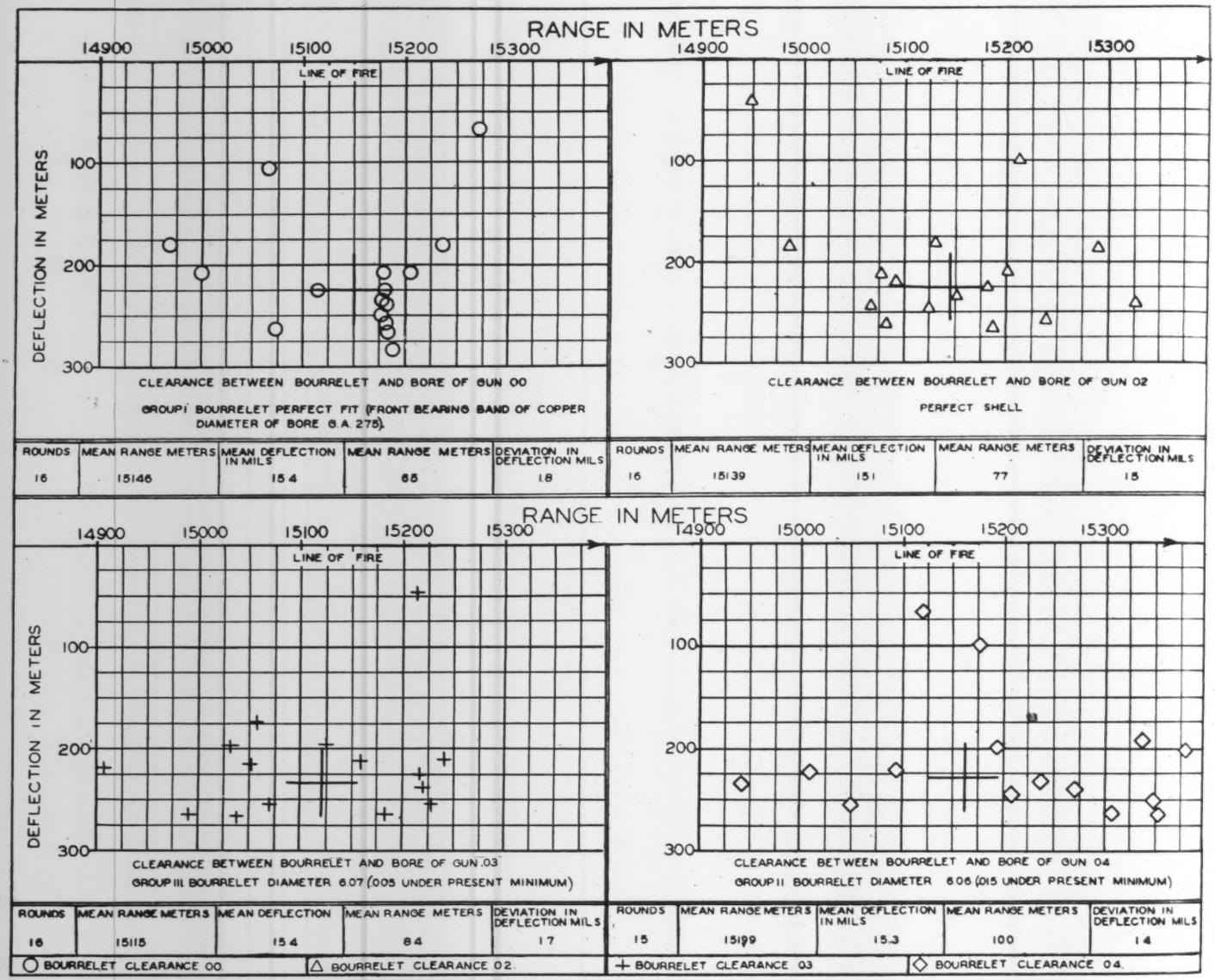

RANGE IN METERS
14900 15000 15100 15200 15300
LINE OF FIRE
DEFLECTION IN METERS
100
200
300
CLEARANCE BETWEEN BOURRELET AND BORE OF GUN 00
GROUP I BOURRELET PERFECT FIT (FRONT BEARING BAND OF COPPER DIAMETER OF BORE 6 A 278).
CLEARANCE BETWEEN BOURRELET AND BORE OF GUN 02
PERFECT SHELL
ROUNDS | MEAN RANGE METERS | MEAN DEFLECTION IN MILS | MEAN RANGE METERS | DEVIATION IN DEFLECTION MILS
16 | 15146 | 154 | 65 | 18
ROUNDS | MEAN RANGE METERS | MEAN DEFLECTION IN MILS | MEAN RANGE METERS | DEVIATION IN DEFLECTION MILS
16 | 15139 | 151 | 77 | 15
RANGE IN METERS
14900 15000 15100 15200 15300
LINE OF FIRE
DEFLECTION IN METERS
100
200
300
CLEARANCE BETWEEN BOURRELET AND BORE OF GUN 03
GROUP III BOURRELET DIAMETER 607 (005 UNDER PRESENT MINIMUM)
CLEARANCE BETWEEN BOURRELET AND BORE OF GUN 04
GROUP II BOURRELET DIAMETER 606 (015 UNDER PRESENT MINIMUM)
ROUNDS | MEAN RANGE METERS | MEAN DEFLECTION | MEAN RANGE METERS | DEVIATION IN DEFLECTION MILS
16 | 15115 | 154 | 84 | 17
ROUNDS | MEAN RANGE METERS | MEAN DEFLECTION IN MILS | MEAN RANGE METERS | DEVIATION IN DEFLECTION MILS
15 | 15199 | 153 | 100 | 14
○ BOURRELET CLEARANCE 00.
△ BOURRELET CLEARANCE 02.
+ BOURRELET CLEARANCE 03
◇ BOURRELET CLEARANCE 04.

FIG. 6 BOURRELET FIT

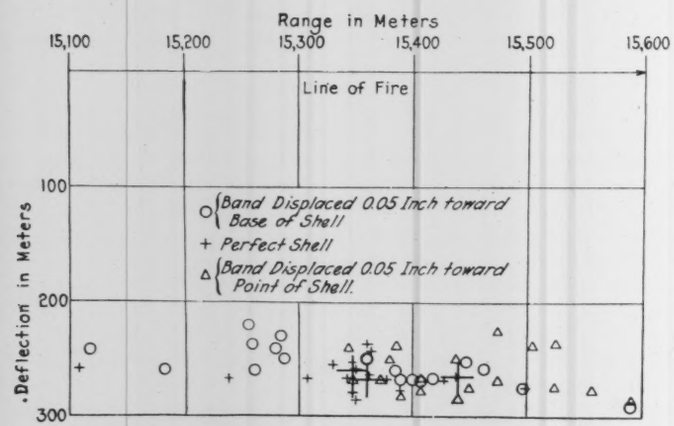


FIG. 5 COMPOSITE GROUPING SHOWING EFFECT OF VARYING DISPLACEMENT OF BAND TO FRONT OR REAR, SUCH AS WOULD BE CAUSED BY WIDE TOLERANCES IN LOCATION OF BAND

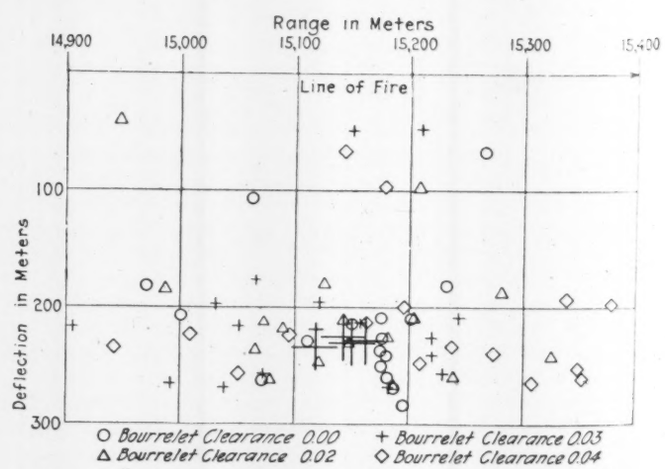


FIG. 7 COMPOSITE GROUP SHOWING EFFECT OF BOURRELET FIT

the band located in the same place, but if the projectiles having different locations of band were mixed in a manner which would result from permitting a large tolerance in the distance from the base of projectile to the band, the dispersion would be greatly increased.

In Fig. 4 are shown the range and dispersion of three groups of shell: (1) With the band displaced 0.05 in. toward the base of the shell; (2) with the band in the normal position shown on the approved design; and (3) with the band displaced 0.05 in. toward the front of the shell. All three groups considered individually are fair, although the one with the band displaced toward the rear shows a marked increase of dispersion. However, when these three groups are plotted as a composite one (Fig. 5) such as would actually occur in service if the tolerance were increased, we find the shots rather widely scattered, 8 out of 11 shorter rounds being those in which the band was displaced to the rear, and 5 out of 6 of the longer rounds being with bands displaced forward.

One dimension on a shell which has always been held to a very close tolerance is the diameter of the bourrelet. Fig. 6 shows the range and dispersion of four groups of shell fired under the same conditions. (1) Shows the results obtained by making the bourrelet of copper and a perfect fit in the bore, and (2), (3), and (4) those when the clearances between the bourrelet and the bore are respectively 0.02 in. 0.03 in. and 0.04 in. The effect of the increasing clearance is shown by the variation in the range and dispersion of each group. Considering each group by itself the results are not so bad, although the dispersion of each group does increase with the increased clearance. The large dispersion in a direction at right angles to the line of fire was partially due to a cross-wind changing in intensity during the firing. All groups are affected, and, referring to Fig. 7, we can see at a glance just what may be expected if we accepted and used shell with such wide differences in dimensions.

Another dimension which is always held to close tolerances is the band diameter. Fig. 8 shows the results obtained with three groups of projectiles which had bands machined (1) to the exact dimension of the drawing; (2) 0.005 in. under the minimum dimension shown on the drawing, and (3) 0.01 in. under the minimum. Here each group considered by itself is satisfactory. Fig. 9 shows dispersion of a composite group of these same shell. In this case the composite group does not greatly increase the size of the pattern. The double grouping is due to the firings being interrupted and completed on another day.

A dimension that one would think might be varied within rather wide limits is the base thickness for shell, yet even small variations in this dimension have a marked effect upon the range and dispersion of a projectile, probably because of the effect on the balance of the shell.

Fig. 10 shows the range and dispersion of three groups of shell, namely, (1) those with a minimum base thickness, (2) those with a normal base thickness (exact dimension shown on the drawing), and (3) those with a maximum base thickness. Here again the individual groups are satisfactory, yet the effect of such variations is at once apparent in a composite group (Fig. 11). The thin-base shell show a decided tendency to range short.

These figures also illustrate some of the dangers of permitting repairs to bring projectiles within the limits of the specifications and drawings. To repair light-weight shell the British permitted the use of special base plates which changed at once the location of the band and the base thickness. Another method permitted the plating of the interior or exterior, and, while this method was probably less objectionable, still it did not put the metal back where the design specified and therefore did not tend toward the production of the uniform product which I have shown is so essential to accuracy of fire.

Necessity of Holding Certain Dimensions to Close Tolerances to Obtain Desired Accuracy

The work just outlined is being continued by the Ballistic Section at Aberdeen Proving Ground, and tests and the collection of data so essential to the design of ammunition which will be satisfactory both from a quality and quantity viewpoint are a most important part of our program to completely develop our designs in peace time. I must again call your attention to the fact, however, that our tests at present indicate that there are certain dimensions which must be held to close tolerances to obtain the desired accuracy and, further, that while it is our intention to permit as wide tolerances as possible, we must insist upon certain dimensions being held to tolerances sufficiently small to obtain the desired accuracy. I feel sure that in this work we shall have your hearty coöperation, for I recall with much satisfaction the work of the various shell and fuse committees which were organized during the war. These committees, composed of manufacturers and representatives of the Ordnance Department, were most successful in solving many difficulties which, without their coöperation would have led to further changes in our drawings and specifications, with a still further loss in the accuracy which we are so desirous of maintaining.

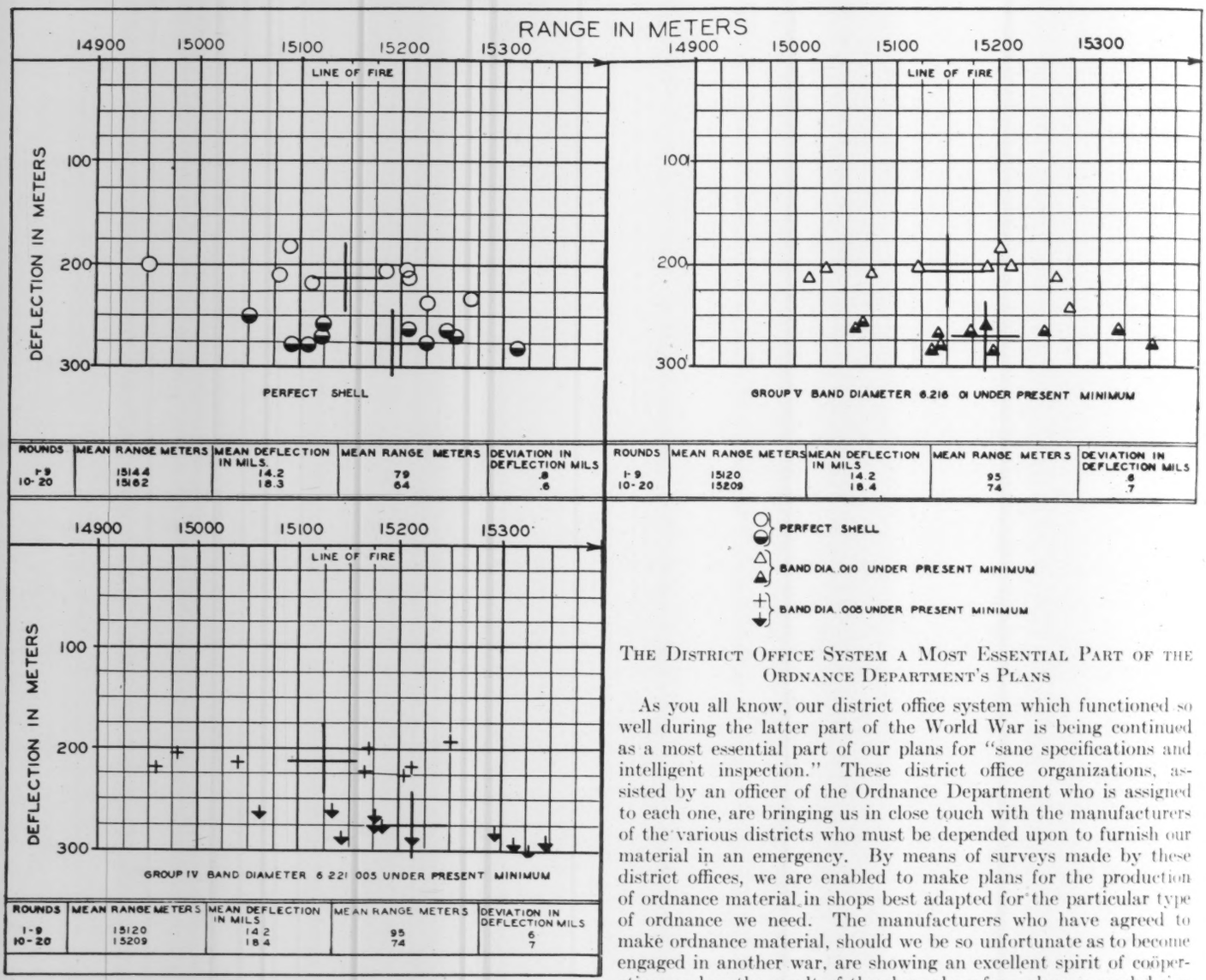
Accuracy of Ammunition Supremely Important

Accuracy of ammunition is always important and cannot be overemphasized. You are all more or less familiar with barrage fire where the advancing infantry follows as closely as practicable the advancing line of bursts of the artillery projectiles. Inaccuracy of ranging results in projectiles falling among our own troops, causing direct casualties and lowered morale, or it requires that the infantry follow at a greater distance, making the barrage less effective and resulting in correspondingly greater casualties from enemy fire. The importance of accuracy was also strikingly brought out in a recent lecture delivered in the Ordnance Office by a field artillery officer who stressed the point that the more accurate we could make the ammunition the less the expenditure that would be required to perform a particular mission. The truth of this statement can be seen from the fact that many rounds of ammunition fired produce no effect because they do not strike sufficiently close to the target to be effective. If by producing more accurate ammunition we can increase the percentage of projectiles that strike within the effective area, we shall have reduced correspondingly the requirements not only of ammunition, but of guns to fire the ammunition, personnel to man the guns, transportation facilities, etc. The possibilities are so evident that it is unnecessary to dwell at length on the importance of accuracy. It is undoubtedly true today, as it was during the World War, that a round of ammunition on the firing line that will shoot is better than a round of the ideal type in the shop. It is also true, however, that a round of good ammunition on the firing line may be worth more than several rounds of the type that can be shot from the gun with safety but with poor accuracy.

Ordnance Department Requirements to Be Liberalized to Facilitate Production

You are likely to conclude from my statements thus far that nothing is being done to liberalize tolerances. As stated at the beginning of the paper, the author's intention has been to cover largely the problem of artillery ammunition, which does not offer a fruitful field for material increases. Among other things, the tests referred to above dealt with the relative effect of a smooth and rough machine finish on the exterior of the shell. The results indicated the undesirability of mixing shell with polished surface with those having a coarse machine finish but indicated no necessity for an extremely fine finish. Some of you will recall that there was considerable controversy on this point because the Russians started out by requiring nickel plating. Other components of ammunition such as fuses are rather delicate mechanisms and offer no great field for liberalizing requirements, but a number of classes of ordnance material do offer such prospects. Artillery guns, carriages, etc., have certain parts such as the recuperators and sight mechanisms which require careful finish, but some of the other parts are less intricate. All requirements are being carefully scrutinized with a view to liberalizing them in such a way as to facilitate production. Tolerances for new designs are being based upon those prescribed in the report of the sectional committee of the American Engineering Standards Committee on Tolerances and Allowances for Machined Fits in Interchangeable Manufacture. In all cases the most liberal tolerances consistent with proper functioning are prescribed.

With the experiments already referred to and others which are under way to determine the basic data for our tolerances, we are also taking steps to correct many of the inconsistencies of our wartime specifications and drawings and thus simplify inspection problems. We have adopted the unilateral system of expressing tolerances and have simplified our drawings by placing the details on



ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS.	MEAN RANGE METERS	DEVIATION IN DEFLECTION MILS
1-9	15144	14.2	79	.8
10-20	15162	18.3	64	.6

ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS	MEAN RANGE METERS	DEVIATION IN DEFLECTION MILS
1-9	15120	14.2	95	.6
10-20	15209	18.4	74	.7

ROUNDS	MEAN RANGE METERS	MEAN DEFLECTION IN MILS	MEAN RANGE METERS	DEVIATION IN DEFLECTION MILS
1-9	15120	14.2	95	6
10-20	15209	18.4	74	7

Fig. 8 Variation in Band Diameter

separate sheets of a size convenient for shop use. Our specifications are now being placed in a standard form and are being coördinated with commercial specifications for the same material as well as with the specifications of other branches of the Government. In making these changes we have profited by the lessons learned during the World War and have made full use of the records which we now have on manufacture and production and which accumulated during that period.

In order to obtain an independent check on the liberality of tolerances prescribed on our own designs, the Gage Section of the Ordnance Department, which is independent of the design sections, has been directed to carefully scrutinize the tolerance on each dimension as the inspection gages are designed, and, while the Gage Section is not authorized to actually increase tolerances, all tolerances thought to be unnecessarily small are brought to the attention of the design sections. It will not be practicable to liberalize the tolerances on all dimensions, but we hope and expect that when a small tolerance is prescribed it will be based as far as practicable on actual necessity for the proper functioning of the material and not on the mere caution of the designer. We are submitting our designs to our manufacturing arsenals for scrutiny as to ease of production. We have also solicited such criticism from private manufacturers, but have as yet not worked out a satisfactory method of obtaining the necessary detailed study. We hope that this may be worked out later perhaps through the medium of our district offices.

The District Office System a Most Essential Part of the Ordnance Department's Plans

As you all know, our district office system which functioned so well during the latter part of the World War is being continued as a most essential part of our plans for "sane specifications and intelligent inspection." These district office organizations, assisted by an officer of the Ordnance Department who is assigned to each one, are bringing us in close touch with the manufacturers of the various districts who must be depended upon to furnish our material in an emergency. By means of surveys made by these district offices, we are enabled to make plans for the production of ordnance material in shops best adapted for the particular type of ordnance we need. The manufacturers who have agreed to make ordnance material, should we be so unfortunate as to become engaged in another war, are showing an excellent spirit of coöperation, and as the result of the shop plans for ordnance work being

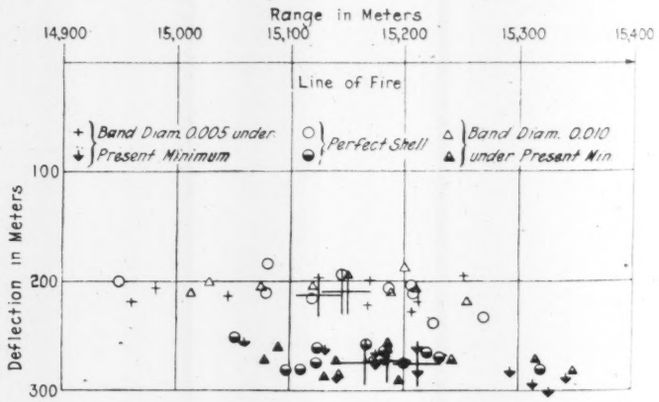
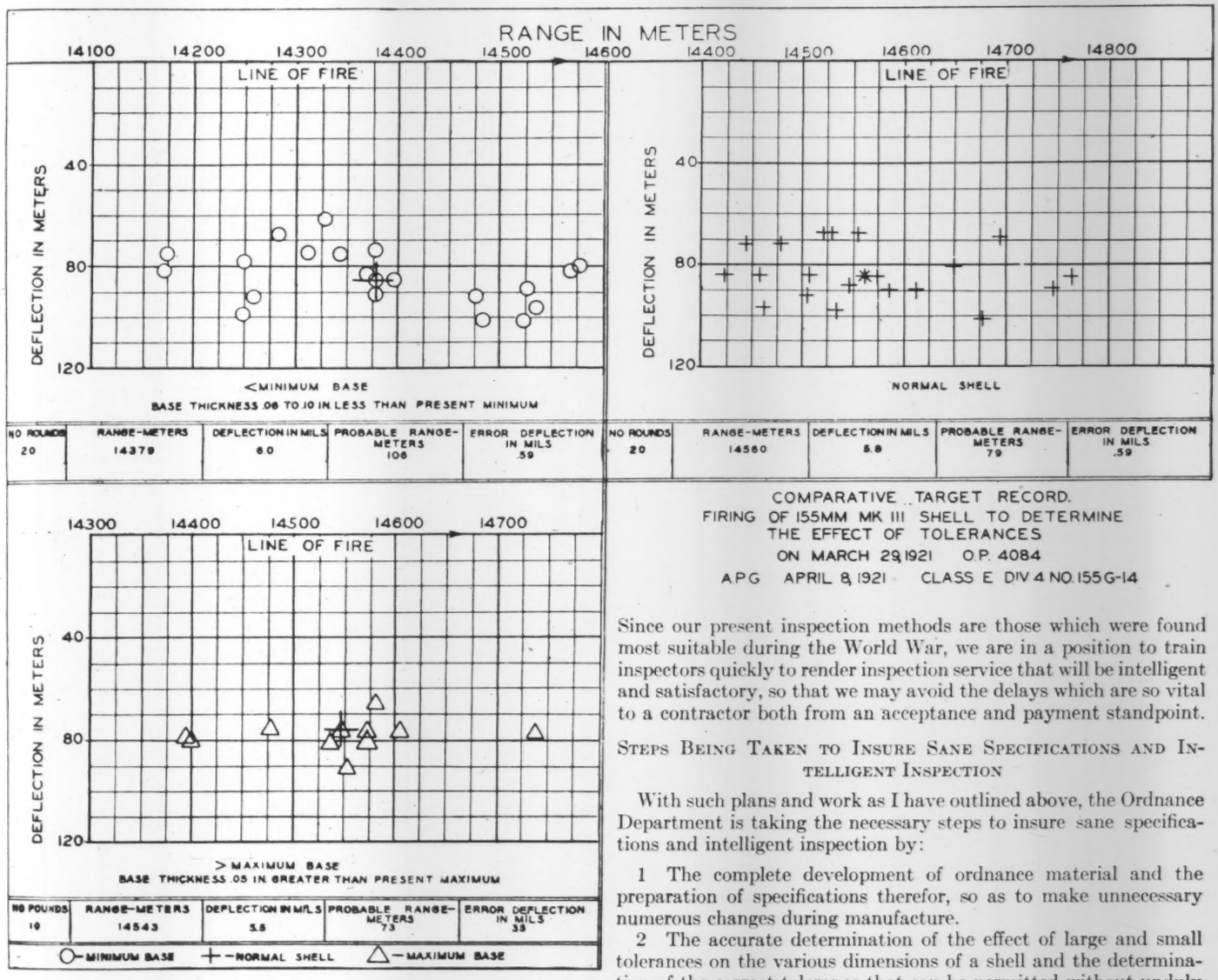


Fig. 9 Composite Group Showing Effect of Variation in Band Diameters

made by them we are receiving much constructive criticism of our drawings, specifications, and inspection methods, all of which tends toward "sane specifications and intelligent inspection," which we so earnestly desire. For the most part, the district organizations are being recruited from experienced men who live in those districts and who know personally most of the manufacturers with whom they will be dealing in an emergency. Many of these men from the district organizations are preparing themselves for


RANGE IN METERS
14100 14200 14300 14400 14500 14600
LINE OF FIRE
DEFLECTION IN METERS
40 80 120
<MINIMUM BASE
BASE THICKNESS .06 TO .10 IN LESS THAN PRESENT MINIMUM
NO ROUNDS 20 | RANGE-METERS 14379 | DEFLECTION IN MILS 6.0 | PROBABLE RANGE-METERS 106 | ERROR DEFLECTION IN MILS .59
14400 14500 14600 14700 14800
LINE OF FIRE
DEFLECTION IN METERS
40 80 120
NORMAL SHELL
NO ROUNDS 20 | RANGE-METERS 14560 | DEFLECTION IN MILS 5.8 | PROBABLE RANGE-METERS 79 | ERROR DEFLECTION IN MILS .59
14300 14400 14500 14600 14700
LINE OF FIRE
DEFLECTION IN METERS
40 80 120
>MAXIMUM BASE
BASE THICKNESS .05 IN GREATER THAN PRESENT MAXIMUM
NO ROUNDS 10 | RANGE-METERS 14543 | DEFLECTION IN MILS 5.8 | PROBABLE RANGE-METERS 73 | ERROR DEFLECTION IN MILS 38
O—MINIMUM BASE +—NORMAL SHELL △—MAXIMUM BASE

COMPARATIVE TARGET RECORD.
FIRING OF 155MM MK III SHELL TO DETERMINE
THE EFFECT OF TOLERANCES
ON MARCH 29, 1921 O.P. 4084
A.P.G. APRIL 8, 1921 CLASS E DIV 4 NO. 155G-14

FIG. 10 EFFECT OF VARIATION IN BASE THICKNESS

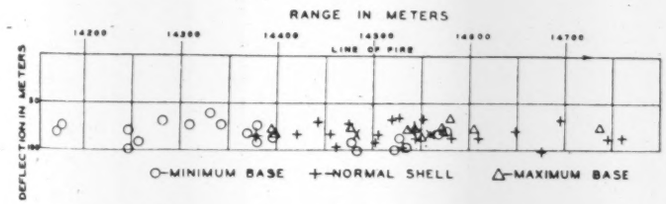


FIG. 11 COMPOSITE GROUP SHOWING EFFECT OF VARIATION IN BASE THICKNESS

ordnance work by spending some time each year at the Proving Ground or at some of the government arsenals, familiarizing themselves with Ordnance Department methods of manufacture, procurement, and inspection.

In addition to these men, who for the most part will act in time of war as district executives in charge of various ordnance activities such as procurement, inspection, and production, we have also enrolled in the reserve corps a selected body of men who are qualified by experience to act as inspectors. To give these men the training in ordnance-inspection methods and processes which will enable them to render intelligent and sound decisions on the points which must arise, even with the best of specifications and drawings, we have established at the various manufacturing arsenals schools for inspectors. At the ammunition arsenals we have equipment and facilities for producing a considerable quantity of ammunition, and in an emergency these facilities would be operated to capacity.

Since our present inspection methods are those which were found most suitable during the World War, we are in a position to train inspectors quickly to render inspection service that will be intelligent and satisfactory, so that we may avoid the delays which are so vital to a contractor both from an acceptance and payment standpoint.

STEPS BEING TAKEN TO INSURE SANE SPECIFICATIONS AND INTELLIGENT INSPECTION

With such plans and work as I have outlined above, the Ordnance Department is taking the necessary steps to insure sane specifications and intelligent inspection by:

1 The complete development of ordnance material and the preparation of specifications therefor, so as to make unnecessary numerous changes during manufacture.

2 The accurate determination of the effect of large and small tolerances on the various dimensions of a shell and the determination of the correct tolerance that can be permitted without unduly affecting the range and accuracy of the projectile.

3 The preparation of specifications and drawings in a standard form, complying as nearly as possible with the best commercial practice.

4 Definite plans for training in ordnance work the district office executives and the chief inspectors, and the selection of carefully trained and experienced men for assignment to this work in an emergency.

5 The preparation of factory war plans in peace time.

6 The establishment of close contacts with manufacturers through our district office system which will eventually place the Ordnance Department designs and specifications on a sound basis for quality and quantity production.

Standardization and Invention

EVERY now and then a voice would be raised in protest or to caution the over-zealous advocates that there is a limit beyond which standardization ceases to be a virtue and becomes a nuisance or even a real brake on progress. Limits set by commercial considerations are often responsible for characteristic developments in engineering and industry. Thus, the standard railroad gage, bridges, tunnels and track clearances fixed years ago are responsible for the present form of railroad locomotives. Great ingenuity has been expended in getting a boiler, coal, and water supply, and steam engine compacted into these limits, and yet efficient enough to develop the power and speed required by the traffic. (Editorial in *Iron Age*, August 5, 1926.)

Permanent Molds for Making Iron Castings

Particulars Regarding a Method of Oil-Flow Cooling That Automatically Maintains the Mold at a Temperature Below That at Which It Will Deteriorate

By H. A. SCHWARTZ,[1] CLEVELAND, OHIO

THE permanent mold has always seemed to be more or less of a mystery to the average man in industry. As a matter of fact, it is quite simple in itself. It differs from the ordinary sand mold in that it has the ability to repeatedly form castings from molten material without breaking down, while the sand mold is destroyed after the first pouring. Foundry procedure—such as gating, venting, feeding, etc.—the proper application of which is so essential to successful casting operations in sand molds, is just as necessary to the successful operation of permanent molds. The real difficulty standing in the way of the commercial development of the permanent mold has been the lack of success in discovering some mold material, or in devising some means, which would enable the permanent mold to withstand the destructive effect upon it of present-day production and metallurgical requirements.

The die casting of the so-called "white-metal" alloys into finished castings came first, and while it was being developed and a degree of success attained commercially, numbers of permanent-mold experiments were being made in the casting of materials of high melting temperature, especially iron. Efforts in this direction paralleled those with die casting. But here the temperature and heat conditions encountered were so much more severe than those obtaining with the die-cast alloys, that the material used for the permanent mold soon deteriorated into worthlessness.

Early Attempts to Cast Iron in Permanent Molds

It is quite probable that the first attempts to cast metals such as iron in permanent molds were made by pouring molten material into a simple block of iron or other metal, gouged out to conform to some pattern shape. This was no doubt the starting point of most experiments with permanent-mold casting. But from this point difficulties were encountered. Without taking into account the usual foundry difficulties due to improper gating, venting, feeding etc., the breaking down of the mold itself after a short period of use presented what seemed to be an insurmountable problem. It was found possible to cast numbers of castings experimentally, but when it was attempted to cast in the permanent mold in quick succession for a substantial casting period—as is necessary for practical operation—the mold heated up. In a very short time, if casting operations were persisted in, the molten material introduced into the mold would fuse to the mold surfaces. Also a criss-cross of cracks, checks, and fissures would appear on the mold surfaces, and pieces of mold material would chip off. As the use of the mold continued, a structural breakdown of the mold material would take place at an accelerating rate, until the mold cavity would be only an approximation of the original pattern. And even before the mold became entirely worthless, the work of producing good castings grew increasingly difficult.

It became apparent at this point in permanent-mold development that until some mold material or some method was found which would make the permanent mold more durable under practical operating conditions, it would be almost impossible to use it successfully for commercial purposes. When this was determined, the attention of investigators centered on devising a permanent mold which would remain serviceable for a sufficient length of time under operating conditions, so that it would be profitable to operate. Various schemes were tried for accomplishing this purpose, most of them falling into two general classes.

Experimental Work on Mold-Surface Coatings

One school of experiment worked on the development of some sort of surface coating for the permanent mold which would serve to protect the mold from the erosive action and heat conditions of the molten material being cast. Presumably they reasoned that inasmuch as the mold structure began to break up at or near the cavity surfaces, it was fair to assume that such deterioration was caused primarily by the destructive effect of the direct contact between the high-temperature molten material being cast and the mold surfaces. They reasoned, then, that a refractory permanent-mold coating would prevent fusing of the molten material to the mold surfaces. They thought that such a coating would retard the entrance of heat into the mold sufficiently so that a large portion of the heat would radiate away; also that the cooling of the forming casting would be retarded to a sufficient degree so that soft, machinable castings would be insured when pouring gray iron. Innumerable experiments were made in attempting to develop a successful mold coating. Tests were made with types of coatings running from thick ones of sand or clay baked to the mold surfaces, to thin ones applied by flame, brush, or air and made up of such materials as silica, chalk, asbestos, clay, carbon, certain metals, and other refractory substances or various mixtures and combinations of them.

Undoubtedly there is a great deal of merit in the application of a suitable coating to a permanent-mold surface. Such a coating will minimize the tendency of molten material to fuse to the mold surfaces when they are very hot; it will retard the entrance of heat into the mold structure, depending on its thickness; and it also is quite helpful in many cases in assuring a soft, gray-iron casting structure by retarding the cooling of the forming casting. But, as has been thoroughly demonstrated, the application of a coating to the mold surfaces does not prevent the structural breakdown of the mold material; it merely tends to slow up the destructive action.

Thick coatings could not be applied successfully because large patches would peel off after a few pouring operations. The remaining coating then had to be removed entirely before applying a new one, if good castings were desired. Thin coatings were more satisfactory in this respect, but they, too, would peel off in a short time. However, the thin coating can be patched, and is the type that has found the most favor. Such coatings are applied to a thickness of only a few thousands of an inch. The use of a refractory coating on the mold surfaces causes a larger quantity of the heat thrown off from the cooling casting to escape directly into the air. But the greater portion of such heat will enter the mold structure, and, if a sufficient time is allowed, will dissipate from the mold through radiation. Even if an adequate time interval between pouring operations were allowed for this purpose, the mold structure would still be subjected to injurious stresses, due to the fluctuations in mold temperature brought on by the successive quantities of heat introduced into the mold.

By taking into consideration the inherent limitations, several permanent-mold methods have been developed along the lines of a coated permanent mold. These methods have been placed on the market and varying degrees of success have been attained in the production of castings of light weight and small cross-section. The commercial production of castings weighing more than a few pounds has not been altogether successful, because of the inability of such methods to handle the heat from heavier castings. Under the most favorable conditions, not more than from 7000 to 10,000 castings can be had from the average mold operating under these methods. The cost of mold replacement and the limitation to the smaller types of castings offset to a large extent the natural advantages obtainable through the use of permanent molds, and are preventing a wider application of permanent-mold methods based solely on the use of a refractory coating.

With certain differences—such as in the mold-coating application—the several methods mentioned embody similar general

[1] Research Engineer, American Foundry Equipment Company. Mem. A.S.M.E.

Presented at a meeting of the Plainfield Section of the A.S.M.E., Plainfield, N. J., May 21, 1926. Abridged.

characteristics. The molds are usually simple gray-iron sections designed to conform to the pattern which is to be cast. As a rule the mold is designed to produce one casting at a pouring, although sometimes, when the casting desired is small, the mold will be arranged to produce two castings at a time. To permit the mold to cool sufficiently after each pouring would make the casting cycle too slow for practical operation. To overcome this objection and still allow sufficient time between pourings for each mold to cool, twelve or more molds are so arranged that they may be operated progressively by one set of men. To operate so many molds independently would be rather awkward and laborious, so a rather complicated turntable mechanism is provided to automatically index and operate the molds arranged thereon as they pass before the different operators. As a rule, three operators are provided for each turntable system; one does the pouring from a guided ladle as the molds pass before him, another sees that the castings are ejected properly and watches the mechanism, while a third inspects the molds and sets cores—if there are any to be set—before the molds are closed for the next pouring.

Each of these permanent-mold methods treats the molds a little differently. One applies a thin coating of silica and silicate o soda—water glass—having a thickness of only a few thousandths of an inch, which is replaced at intervals as it peels off. As a further precaution, a coating of carbon is deposited on the mold surfaces after each pouring operation. Another method employs a coating of zinc which is deposited on the mold surfaces. It is claimed that this coating will prevent the molten material from fusing to the mold surfaces. The injurious effect of zinc fumes on any highly heated iron structure is well known. A carbon coating deposited by flame after each pouring operation is also employed in this method. In fact, the coating of permanent molds with carbon by various means is pretty generally practiced. In this method an attempt is also made to so design the mold section that it will draw the heat from the cooling casting uniformly. In still other permanent-mold methods french chalk and silicate of soda—acting as a binder—are used as the coating. In all of these methods the cooling of the mold during the interval between pourings is further promoted by the use of a current of air directed against the mold cavities.

Experiments in Mold-Surface Cooling

The second school of experiment decided after their initial tests that the problem of permanent-mold deterioration could be solved by establishing some means for keeping the mold temperature at all times below a certain point. Several means for accomplishing this purpose were tried. In some cases the permanent mold was made of a large mass of metal on the theory that the heat introduced into the mold would distribute itself throughout the mold mass and thereby lower the temperature level of the mold. It was claimed by some of those experimenting in this direction that the mold mass should be about one hundred times that of the casting being poured. If the casting being poured weighed 1 lb., then the mold should weigh 100 lb. The greater mold mass was slower in heating up, it is true. But the heat input was constant under the casting conditions, and the mold was soon even hotter than if it had been of the ordinary design. This was so because the proportion of mold mass to mold surface was greater, resulting in a decrease of the radiating surface in proportion to the heat volume. Injurious mold stresses were really intensified, and experiments in this direction resulted in failure. In other cases permanent molds were cooled by the use of a current of air directed against the mold surfaces. As air is a poor heat conductor, its help in cooling the mold is more or less negative and indirect. By setting up air currents near the mold surfaces, radiation of heat from the mold is accelerated; but if the amount of the heat input into the mold is considered they are ineffective as a prompt and efficient means of cooling.

Water was most commonly used in the numerous experiments for cooling permanent molds, but it did not prove successful, as steam pockets formed, and the life of the molds was short.

The Author's Experiments

Several years ago the author became interested in permanent molds for casting materials of the higher melting temperatures. It seemed to him, after a thorough study of the past development in the art, that the crux of the problem had been missed. With the help and encouragement of several men interested his conclusions were incorporated in equipment which was tested for a period of several years under actual foundry conditions and various types of castings produced, ranging from 1 lb. to 35 lb. in weight. These tests conclusively proved that the process would substantially prolong the serviceable life of the permanent mold and was superior to anything known before. About a year or so ago the patents covering the process were trusteed and arrangements completed for making it known to the trade.

The problem of preventing deterioration of the mold structure under operating conditions is essentially one of heat and temperature. This was thoroughly proved in the course of the author's work and the crux of the problem was found to be not as much in the high temperature as it was in the widely fluctuating uneven temperatures attained in different portions of the mold structure. This temperature condition subjected the mold structure to violently changing contractions and expansions, and no material could withstand the resulting structural stresses for any length of time.

It was determined that if the following control of the mold temperature could be maintained, then the greater part of the difficulty would be overcome:

1 Stabilize the temperatures obtaining at different parts of the mold structure around a common point within as close limits as possible;

2 Establish a suitable predetermined mold temperature; and

3 Maintain this established mold temperature within as close limits as possible during and between pouring operations throughout the entire casting period.

It was realized, of course, that it would be impractical to maintain such a control exactly, but even if it were only approximate, it was believed that conditions would be improved to such an extent that deterioration of the mold material would be largely overcome.

Suitable stabilized temperature ranges for each of the mold sections are determined from the character of the material being cast, the type of casting structure desired, and the distribution of material in the casting to be made. If cast iron is to be poured, the temperature range of each of the several mold sections is stabilized at a higher or lower zone to retard or to accelerate the cooling of the casting, depending on whether machinable or chilled surfaces are required.

Generally speaking, a permanent mold for casting thinner sections should be stabilized at a higher temperature than would be required for one for casting thicker-sectioned castings. A thin-sectioned casting has more surface exposed to the mold in proportion to its mass than a thicker-sectioned one, and by varying the stabilized mold temperature to suit the character of casting the rate of cooling can be controlled to a certain extent. Whenever possible where both thin and thick sections occur in the same casting, the mold is made with varying wall thickness, and fins, etc. are sometimes added to increase the cooling action at certain places. This helps in abstracting heat at proper rates to produce uniform structural conditions in the casting.

Automatic Control of Temperature of Permanent Molds

An ideal means for accomplishing the exact temperature control of the permanent mold is one that will free the mold of all the surplus heat when, where, and as it is abstracted by the mold from the successive charges of molten metal. Of course, under the conditions only an approximation of this could be effected. The heat-absorbing medium had to be one that was available at a reasonable cost, and one that could be easily handled without the need for complicated or costly equipment. It was determined that the ideal heat-absorbing medium had to be in a stable liquid state at the operable temperature conditions obtaining in the permanent molds. It was found that the medium must be in a stable liquid state somewhere between the boiling point and the critical temperature of water, i.e., between about 200 and 700 deg. fahr. Of course, between these limits certain materials were better suited than others, but there were only a very few to choose

from. Water was quickly discarded because of its general physical and thermal instability at the operating temperatures. It also seemed to have too sharp a chilling action on the mold material. Gases were also discarded for the same reason of instability, but with them the absorption of heat took place too slowly.

A number of metallic and non-metallic materials having the approximate characteristics desired were tried, including several fusible alloys, mercury, tin, and various oils and compounds. Although many of these materials seemed more or less satisfactory, final choice centered on the use of certain oils, chiefly on account of their availability and ease in handling. The mold walls are made as thin as is possible and yet not have them subject to breakage under the mechanical stresses. In many cases they are less than $\frac{1}{2}$ in. thick. One or more fluid chambers are arranged at the back of each mold section, through which the heat-absorbing medium is circulated in proximity to the mold-cavity walls by means of a pump. Valves are arranged to throttle the flow of fluid through the molds in response to the temperature changes in the mold. If the mold temperature rises, the flow is increased. If the mold temperature falls—as it will between pouring operations—the fluid flow is decreased, and is stopped altogether when the mold temperature drops below a certain point. As the oil remains in a stable liquid condition during its course through the molds, it has a tendency to stabilize within close limits the temperature spread over the mold surfaces and structure. To insure a regulation that is both sensitive and prompt in stabilizing the mold-temperature conditions, the heat-absorbing fluid flow is controlled by thermostatic devices governed from the temperature of the mold sections. In this way a predetermined mold-temperature range may be established and maintained throughout an entire casting period. The presence of a stable liquid in contact with the mold walls automatically equalizes to a large extent the temperature conditions throughout the mold structure. In the majority of cases the mold temperature is stabilized at levels between 350 to 750 deg. fahr. The range at any one point on the mold structure does not exceed 75 to 100 deg., although it is satisfactory to allow variations of as much as 100 deg. or more from the predetermined point. In tests on a mold for casting an article weighing over 35 lb. the temperature variation at one point close to the casting surface was found to be less than 50 deg. fahr. over an extended casting period.

In many cases, if the conditions require, the inner faces of the mold walls are arranged with fins, etc. to offer a maximum cooling surface to the heat-absorbing medium, and also to break up the oil flow and help create a turbulent flow of the oil as it circulates through the mold chambers. The size, shape, and weight of the casting and the character of material being cast influence the design of the mold as regards the number of sections, parting, gating, venting, and feeding, as well as other important features of permanent-mold construction. Although general design and constructional data have been determined, each new casting presents certain special problems of mold construction. Naturally, the process is profitable or commercially suitable only where large quantities of standardized castings are required. Of course there are exceptions to this rule, but generally the use of the process is not warranted where only a relatively small number of parts are required.

Advantages of System Described

Because of the means provided for carrying off promptly all surplus heat accumulating in the mold from the successive pourings, there is no limitation to the speed at which the molds may be operated. Also, for the same reason, there is no limitation to the size of casting which may be poured. Consequently it is not necessary to provide a prolonged time interval between pourings by operating a large number of molds in one system. This does away with the necessity of an expensive and complicated automatic mechanism for handling a number of molds in one operating system. By arranging the mold to cast a number of castings from each pouring, a few molds operating independently and handled by two men will more than equal the producing capacity of twelve or more molds of the refractory-coating type operating on an automatic turntable device. For convenience, about six of the molds are placed together in one operating battery. They are absolutely independent of each other in operation and may be arranged to cast different types of castings if desired. Simple means are provided for connecting or disconnecting the fluid service, so that the units may be exchanged or taken away without interruption to the casting operation. If full value from the process is to be obtained, a continuous means for obtaining molten metal must be available. To give an idea of the production possibilities, the author would say that it usually requires about five to ten seconds to fill a mold with molten material. The casting is ready to be ejected from the mold in about ten to twenty seconds. The mold can be prepared again for a new pouring operation in much less than 30 seconds, even if there is a core to set.

Shrinkage and Strength of Permanent-Mold Gray-Iron Castings—Life of Molds

In the discussion following the presentation of the paper, interesting particulars were brought out regarding the shrinkage, strength and finish of permanent-mold gray-iron castings and on the life of the molds.

The shrinkage of a gray-iron casting made in the oil-cooled permanent mold, according to the author's experience, was approximately one-half of what the shrinkage of the same piece would be when made in a sand mold. The reason for this smaller shrinkage being questioned by some of the experienced foundrymen present, the author replied that it was probably due to the fact that a part of the iron "set" very quickly and before the mold was entirely filled, with the result that the metal last poured supplied the shrinkage material in the same way as the riser in a sand casting was supposed to do, but in a more effective manner. He further stated that the question of suitable gates and risers was just as important and required as great study and care for the production of satisfactory castings as in the case of sand molds.

Permanent-mold castings, the author stated, had a greater strength than sand-mold castings due to the greater density of the iron. Actual experience with test pieces had shown this beyond doubt, and in many cases their strength had exceeded that of the sand castings by from 40 to 50 per cent.

The life of a mold, the author said, should exceed 100,000 castings, as in some molds over 30,000 castings had already been produced with practically no signs of wear or attack by the molten iron on the mold.

Castings which were exhibited by the author at the meeting had an excellent surface finish.

THE problem of production of iron from ilmenite ores is complicated by the desire to recover titanium for pigmentary or other purposes, the latter metal being usually recovered in the form of titanium oxide. The work described here was done by the Hydrometallurgical Laboratory of the Department of Mines, Dominion of Canada. The ores used contained 31.6 per cent of titanium oxide and 40.5 per cent of iron. The material was first crushed to minus 80 mesh and then reduced in crucibles to spongy iron. This was leached at various temperatures and the leach liquor, after the removal of such impurities as copper, lead, nickel, and cobalt, was subjected to the electrodeposition process in a diaphragm cell having a steel mandrel of 1 sq. ft. plating surface as cathode and two graphite anode plates.

Current densities around 100 amperes per sq. ft. have proved satisfactory. The presence of nickel, copper, or cobalt in the electrolyte in quantities less than 0.02 gr. per liter has no apparent effect on the deposit. Larger quantities tend to cause cracking or stripping of the deposit. Lead in quantities above 0.002 gr. per liter gave trouble. Deposition at 70 deg. cent. or lower is attended with nodular formations, and generally uneven deposits.

It is claimed that the process is complete and commercially profitable. The concentrations of titanium will, it is said, be produced so cheaply that it will be possible to make a white pigment that can be placed on the market at prices competitive with zinc oxide, and possibly also with lithopone. (R. H. Monk and R. J. Traill in a paper presented at the Annual Dominion Convention of Chemists, Montreal, June, 1926; abstracted through *Canadian Chemistry and Metallurgy*, vol. 10, no. 6, June, 1926, pp. 137–139.)

Stresses on Bolts—Nut Dimensions—Wrench Design

By GEO. S. CASE,[1] CLEVELAND, OHIO

THE purpose for which the following study was made was to gather data concerning the resistance of bolts, nuts, and wrenches to stress that might be of value in their design, selection, and use. The first section deals with the relation of the torque applied to a nut and the resulting strains in the bolt, together with certain related data useful in designing and using bolts, and other threaded parts. The second section has to do with the strength and torque transmitting capacities of nuts, while the third is devoted to the design of wrenches in relation to their effective use.

The Stresses on a Bolt

In most cases the efficiency of a bolt depends upon the care used in tightening it, and in deciding how a bolt should be tightened it is necessary to consider what the stresses on it will be. These are of four different classes depending upon the use to which the bolt is put, namely,

1 A bolt holding a part which is locked by other means.
2 A bolt holding a part which is held in place by the grip of the bolt only.
3 A bolt with a compressible substance, such as a spring or a compressible packing, between the two surfaces it is holding together.
4 Bolts subject to shear.

In the first and fourth classes it is necessary to tighten the bolts to such an extent that they will be held firmly in place and will not work loose.

The larger number of bolts are used where they come under the second class. Here the efficiency of the bolt depends entirely upon the initial tightening. If a load tends to separate the two parts held in contact, no additional stress is put upon the bolt until the load equals the original tension in the bolt, but as soon as this tension is exceeded the two parts separate. Connecting-rod bolts on the crank end are in this class, and in these if the initial tension applied by the nut is exceeded, the cap separates from the rod.

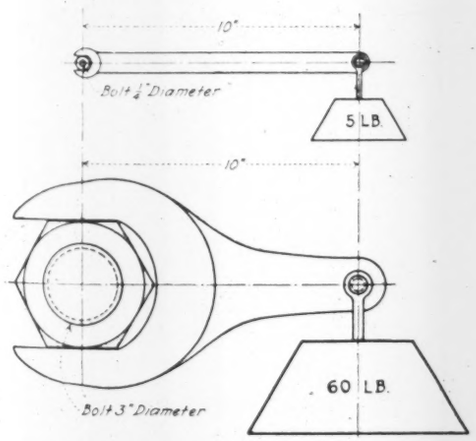


Fig. 1 Torque to Put 1000 Lb. Pull or Grip on Bolts of Different Diameters

In class 3 the live load is immediately added to the initial tension in the bolt, a conspicuous example of this class being the bolts holding a cylinder head on against compressible packing. In this case any live load produced by steam or hydraulic pressure inside the cylinder is immediately added to the initial tension in the bolts, so that it is important that such bolts should not be tightened to greater initial tension than necessary.

Track bolts are a combination of classes 1 and 2. In order to compute the stresses in them it is necessary to know the angle at the top and the bottom of the splice bar and the coefficient of friction between the splice bar and the rail. Tests have indicated that additional tightening of the nut very greatly strengthens the rail joint, considering the latter as a girder.

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In addition to the foregoing, bolts are subjected to torsional stress due to the friction between the nut and the bolt.

Whenever the initial tightening stress is the important one, it is necessary to have some knowledge regarding the relation between the torque applied and the initial tension in the bolt.

When a reasonably well-lubricated bolt (coefficient of friction = 0.15) is tightened, the relation between the torque on the nut and

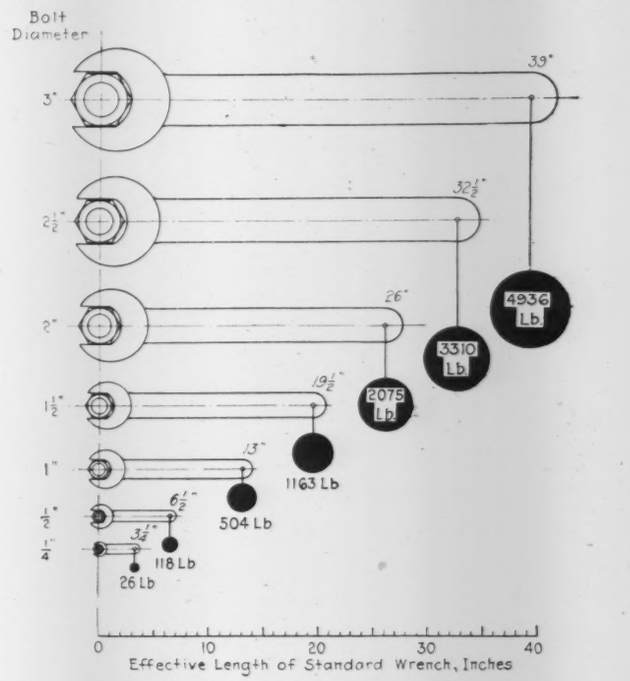

Fig. 2 Pounds Pull to Produce a Stress of 60,000 Lb. per Sq. In. in Bolt

the tensile stress in the bolt is given by the equation

$$Pl = RDT$$

where P = pull at end of wrench, lb.
l = effective length of wrench, in.
D = nominal or outside diameter of the bolt, in.
T = tensile stress on bolt, lb. per sq. in., and
R = factor given in Table 3.

The actual value of R varies from 0.205 to 0.220, but for ordinary calculations it may be taken as 0.200, so that the formula becomes

$$Pl = 0.2DT \text{ or } T = \frac{5Pl}{D}$$

With a constant torque the grip of a bolt is inversely proportional to its outside or nominal diameter. Stating this another way, the torque required to produce a given grip varies directly as the nominal diameter of the bolt.

The importance of this relation is shown by reference to Figs. 1 and 2. Fig. 1 shows the pull which must be exerted at the end of a 10-in. wrench in order to produce a grip of 1000 lb. in a 3-in. bolt, compared with that which must be exerted to produce the same grip in a ¼-in. bolt. Fig. 2 shows the necessary pull on standard-length wrenches in order to produce stresses of 60,000 lb. per sq. in. in bolts of different diameters.

Where the stress (S) on the material is important rather than the pull on the individual bolt, the formula becomes approximately

$$Pl = 0.1D^3S \text{ or } S = \frac{10Pl}{D^3}$$

This formula may be as much as 15 per cent in error, and the figures in the tables should be used if available, data for American National Coarse or U. S. Standard threads being given in Table 1, and for American National Fine or S.A.E. threads in Table 2.

The figure of 60,000 lb. per sq. in. is assumed arbitrarily since it is near the ultimate strength of low-carbon-steel bolts without special heat treatment. It is a figure which can be used as a basis for computing the torque required to load alloy-steel bolts and heat-treated carbon-steel bolts of greater ultimate strength, since the torque required will be proportional to the grip of and the stress on the bolt. However, the elastic limit might be a safer figure to use, and should be taken at about one-half the ultimate strength.

It is apparent that, since the torque required to load a bolt at a given stress per square inch varies as the cube of the diameter, standard wrenches produce a relatively high strain in bolts of small diameter and an extremely low one in bolts of large diameter.

It must be appreciated that the torque necessary to produce a grip or pull in a bolt varies directly as the diameter of the bolt. This is of importance when applied to many threaded parts. For instance, if a screw is used as a jack it is clear that the smaller the diameter the more efficient it will be.

Again, the strength of a rail joint depends on the grip of the bolts,

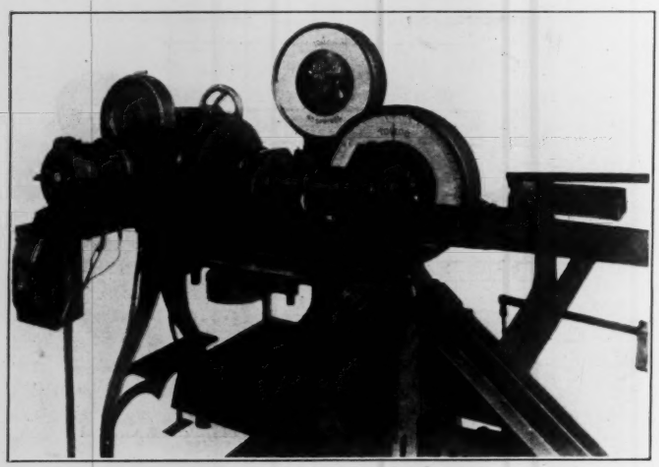

FIG. 3 MACHINE MEASURING SIMULTANEOUSLY THE TENSILE PULL ON A BOLT AND THE TORQUE TRANSMITTED TO ITS NUT

and it should be considered whether more small bolts of ordinary steel or smaller bolts of higher-strength steel should be used so as to make it possible for the man tightening them to load them to their full capacity.

The conditions which must be considered in the tightening of bolts fall roughly into three classes.

In small sizes the danger of breaking the bolt is the principal one encountered. The torque necessary to break a 1/4-in. bolt of 60,000 lb. per sq. in. ultimate strength is 87 in-lb. This requires little pull, even when using a short wrench. Bolts 5/8 in. in diameter are about the smallest that are not readily broken with a standard wrench if the usual low-carbon steel is employed in their manufacture.

In the medium group, from 5/8 in. to 1 in. or slightly larger, it is possible to apply a torque that will utilize the full strength of the bolt, but not with standard-length wrenches. This is the reason why track bolts are usually tightened with longer wrenches, and why such wrenches are frequently used in car-construction and other work. A careful study of the tables given should indicate the kind of wrench to employ and the amount of pull desirable, and should make it possible to select the best bolt for the work.

Large bolts (over 1 in. diameter) require so much torque to tighten them to a point where their strength is efficiently utilized that it would seem wise to consider employing a larger number of smaller bolts of the same or a higher tensile strength, if initial tightening is necessary—as is usual.

The data in the tables were developed mathematically[2] and have been checked experimentally by means of a machine designed by the author and built by The Lamson & Sessions Company. This machine measures simultaneously the tensile pull on a bolt and the torque transmitted to its nut.

With this machine, a photograph of which is shown in Fig. 3, a number of men were requested to tighten a nut until they considered it snug, and then to pull the nut as tight as they could. The dial on the scale was covered. As might be expected, the results were erratic. The pull on standard commercial wrenches varied from 25 to 50 lb. on a 1/4-in. bolt and from 50 to 180 lb. on a 1-in. bolt. The position of the nut and wrench was probably as good as

[2] The mathematical development of these data is on file at the headquarters of the Society, and may be consulted by those interested.

TABLE 1 DATA ON AMERICAN NATIONAL COARSE THREAD SYSTEM

Diameter of bolt, in.	Threads per inch	Torque on nut for 1000 lb. tension in bolt, in-lb.	Torque on nut for 60,000 lb. per sq. in. stress in bolts, in-lb.	Effective length of standard commercial wrenches, in.	Pull on standard commercial wrenches to produce stress of 60,000 lb. per sq. in., lb.
1/4	20	55	87	3.25	26
5/16	18	68	178	4.06	44
3/8	16	81	319	4.88	65
7/16	14	93	501	5.69	88
1/2	13	107	771	6.50	118
9/16	12	121	1,124	7.31	153
5/8	11	134	1,538	8.13	189
3/4	10	160	2,730	9.75	280
7/8	9	186	4,371	11.4	383
1	8	212	6,554	13.0	504
1 1/8	7	239	9,295	14.6	636
1 1/4	7	264	13,086	16.3	802
1 1/2	6	315	22,686	19.5	1,163
1 3/4	5	368	35,828	22.8	1,571
2	4 1/2	420	53,961	26.0	2,075
2 1/4	4 1/2	471	78,762	29.3	2,688
2 1/2	4	523	107,589	32.5	3,310
2 3/4	4	573	145,761	35.8	4,071
3	3 1/2	623	192,514	39.0	4,936

TABLE 2 DATA ON AMERICAN NATIONAL FINE THREAD SYSTEM

Diameter of bolt, in.	Threads per inch	Torque on nut for 1000 lb. tension in bolt, in-lb.	Torque on nut for 60,000 lb. per sq. in. stress in bolt, in-lb.	Effective length of standard commercial wrenches, in.	Pull on standard commercial wrenches to produce stress of 60,000 lb. per sq. in., lb.
1/4	28	53	98	3.25	30
5/16	24	67	199	4.06	49
3/8	24	79	356	4.88	72
7/16	20	93	565	5.69	99
1/2	20	105	864	6.50	133
9/16	18	118	1,234	7.31	168
5/8	18	128	1,688	8.13	207
3/4	16	156	3,015	9.75	309
7/8	14	182	4,768	11.4	418
1	14	206	7,276	13.0	559
1 1/8	12	233	10,325	14.6	707
1 1/4	12	257	14,202	16.3	871
1 1/2	12	307	25,102	19.5	1,287

TABLE 3 BOLT DATA

Diameter, in. D	Turns per inch N	Pitch diam. basic, in. E	Min. root diam., in. K	Helix angle	$L = \frac{1}{N}$ L	$\frac{D}{K}$	Min. K^2	J	R	G
1/4	20	0.2175	0.1814	4°11′	0.0500	1.38	0.033	1.11	0.220	0.98
5/16	18	0.2764	0.2362	3°40′	0.0555	1.32	0.056	1.10	0.217	1.00
3/8	16	0.3344	0.2893	3°24′	0.0625	1.30	0.084	1.09	0.216	1.00
7/16	14	0.3911	0.3398	3°20′	0.0714	1.29	0.116	1.09	0.212	1.01
1/2	13	0.4500	0.3949	3° 7′	0.0769	1.27	0.156	1.08	0.215	1.02
9/16	12	0.5084	0.4486	2°59′	0.0833	1.26	0.201	1.08	0.214	1.02
5/8	11	0.5660	0.5010	2°56′	0.0909	1.25	0.251	1.08	0.214	1.03
3/4	10	0.6850	0.6137	2°40′	0.1000	1.22	0.377	1.07	0.213	1.04
7/8	9	0.8028	0.7237	2°31′	0.1111	1.21	0.524	1.06	0.212	1.05
1	8	0.9188	0.8300	2°29′	0.1250	1.21	0.689	1.06	0.212	1.05
1 1/8	7	1.0322	0.9309	2°31′	0.1429	1.21	0.867	1.06	0.212	1.05
1 1/4	7	1.1572	1.0559	2°15′	0.1429	1.18	1.12	1.05	0.211	1.06
1 1/2	6	1.3917	1.2734	2°11′	0.1667	1.18	1.62	1.05	0.210	1.06
1 3/4	5	1.6201	1.4786	2°15′	0.2000	1.18	2.19	1.05	0.210	1.06
2	4 1/2	1.8557	1.6986	2°11′	0.2222	1.18	2.89	1.05	0.210	1.06
2 1/4	4 1/2	2.1057	1.9486	1°55′	0.2222	1.16	3.80	1.05	0.209	1.07
2 1/2	4	2.3376	2.1612	1°57′	0.2500	1.16	4.67	1.05	0.209	1.07
2 3/4	4	2.5876	2.4112	1°46′	0.2500	1.14	5.83	1.04	0.209	1.08
3	4	2.8376	2.6612	1°36′	0.2500	1.13	7.08	1.04	0.208	1.08
1/4	28	0.2268	0.2014	2°52′	0.0357	1.24	0.041	1.07	0.214	1.03
5/16	24	0.2854	0.2560	2°40′	0.0417	1.22	0.066	1.07	0.213	1.04
3/8	24	0.3479	0.3185	2°11′	0.0417	1.18	0.101	1.05	0.210	1.06
7/16	20	0.4050	0.3697	2°15′	0.0500	1.18	0.137	1.05	0.211	1.06
1/2	20	0.4675	0.4324	1°57′	0.0500	1.16	0.187	1.05	0.209	1.07
9/16	18	0.5264	0.4873	1°55′	0.0555	1.15	0.238	1.05	0.209	1.07
5/8	18	0.5889	0.5498	1°43′	0.0555	1.14	0.302	1.04	0.205	1.08
3/4	16	0.7094	0.6656	1°36′	0.0625	1.13	0.443	1.04	0.208	1.08
7/8	14	0.8286	0.7786	1°34′	0.0714	1.12	0.606	1.04	0.208	1.09
1	14	0.9536	0.9036	1°22′	0.0714	1.11	0.817	1.03	0.206	1.09
1 1/8	12	1.0709	1.0127	1°25′	0.0833	1.11	1.03	1.03	0.207	1.09
1 1/4	12	1.1959	1.1377	1°16′	0.0833	1.10	1.29	1.03	0.206	1.10
1 1/2	12	1.4459	1.3877	1° 3′	0.0833	1.08	1.93	1.02	0.205	1.11

could be expected in the case of the smaller bolts, but not as good to put all a man's weight on the wrench on a 1-in. bolt as it would be on many machines or on track bolts.

Usually it will be the maximum pull that is important on small bolts, where the principal danger is that of breaking them. On large bolts, as for instance, track bolts, probably a pull of from 50 to 75 lb. will be put on by a man using reasonable effort, and one of 200 lb. under the best conditions. The author has checked these figures in a number of ways, and with a number of different kinds of experiments, and while there is a wide variation in them they may be taken as representing the limits for ordinary conditions. With a pipe slipped on the end of a wrench and his shoulder under it, a man can exert a force of considerably over 200 lb. It is the author's belief that bolts $^1/_4$ in. in diameter and smaller should be designed so that a pull of 50 lb. on the wrench will not break them. Most men can put about a 50-lb. pull on a 4-in. wrench on a $^1/_4$-in. bolt. Men using wrenches 10 in. or 12 in. long will usually exert a pull of about 100 lb. if they make a decided effort to get the bolt tight. If a strong man applies himself and is in a good position to put his weight into the effort, he can pull approximately 200 lb.

A Study of Nut Dimensions

The following paragraphs deal with hexagon nuts, considering them with particular reference to their ability to transmit torque.

The quantity manufacture of nuts and bolts is of comparatively recent origin, dating back only to about 1860. In the early years it was quite customary to make nuts of various widths up to twice the diameter of the bolt. But little attempt was made to standardize this dimension until 1876, when the so-called Sellers or Franklin Institute, or, as it has been still better known, the U. S. Standard, series of nut sizes was established. At that time nuts were made of iron or welded scrap, which was weak in one direction, so that the dimension of $1^1/_2$ diameters plus $^1/_8$ in. wide by one diameter thick was undoubtedly necessary and well balanced. Up to 1904 or thereabouts heavier nuts than the United States Standard were frequently used. Between 1900 and 1910 steel became cheaper than iron or welded scrap so that steel nuts gradually replaced iron ones, and for the last fifteen years there have been none heavier than U. S. Standard in common use. Until recently a considerable number of iron nuts were still being made, but at the present time commercial nuts are nearly all steel. The workmanship on bolts and nuts has also improved greatly.

Lighter steel nuts will now do the work of the U. S. Standard nuts of 1876, and we find several lighter series of nuts in use. Prominent among these are the S.A.E., the Bolt Manufacturers', and the Agricultural Implement Manufacturers' standards. In addition many individual concerns use lighter nuts than the U. S. Standard in order to save material or to take advantage of the smaller size of the nuts in inaccessible places where they are hard to turn.

Experiments and the experience of users indicate that the limiting factor in the outside dimensions of a nut is its ability to transmit torque rather than its strength.

The author has made numerous tests by sawing nuts more than half way through and tightening them under all kinds of conditions. He has never found anything to indicate that any steel nut in use today is so light that there is a question of strength of side wall.

A nut whose thickness is $87^1/_2$ per cent of the bolt diameter is entirely satisfactory except in few cases. For many uses a thickness of 75 per cent of the bolt diameter meets the requirements as far as the strength of the thread is concerned.

A Sectional Committee on Bolt, Nut, and Rivet Proportions, organized under the procedure of the American Engineering Standards Committee has studied the matter carefully and its report[3] presents a series of nuts and bolts proposed with the idea that they will entirely replace the S.A.E., the Bolt Manufacturers', and the Agricultural Machinery series, and the U. S. Standard nut in nearly all cases.

The author has not considered the matter of bearing surface under the nut and whether a nut loosens more readily if it is thinner. It is his opinion that there are very few cases where the nuts given in these tables will not afford ample bearing surface. The fact that today 90 per cent of the square nuts used are of smaller dimensions than the U. S. Standard, would indicate that the bearing surface of hexagon nuts of the same dimensions should be satisfactory. It has never been proved to the author's satisfaction that thick nuts hold tight better than thin ones, but it is standard railroad practice to use them.

One reason for the importance of the ability of a nut to transmit the torque of the wrench without rounding its corners is that failure at this point is more annoying to the user than failure in any other way. Bolt manufacturers find that the users of nuts are seriously bothered if their wrenches go around the corners of the nuts. The next most serious failure to the user is to have the threads strip, so there should be sufficient excess capacity in the thread and at the nut corners so that the bolt will break before either of the other things happen. Bolts designed in this way are more satisfactory in use than the theoretically perfectly balanced bolt in which all three forms of failure would happen at the same time.

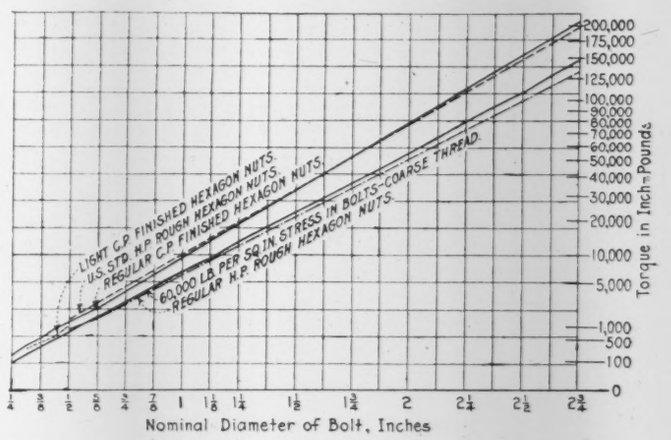

FIG. 4 TORQUE IN INCH-POUNDS AT WHICH WRENCH WILL GO AROUND CORNERS OF NUTS COMPARED TO TORQUE REQUIRED TO LOAD BOLTS

TABLE 4 WRENCH TORQUE AND TORQUE TRANSMITTED BY HEXAGON NUTS

Diam. of bolt, in.	Torque to Load Bolts to 60,000 Lb. per Sq. In. Fine thd.	Coarse thd.[1]	Torque in Inch-Pounds Transmitted by Hexagon Nuts: Light C.P. finished[1]	Reg. C.P. finished[1]	Reg. C.P. rough	Reg. H.P. rough[1]	U.S.S. C.P. finished	U.S.S. H.P. rough[1]
			Brinell Hardness: 140	140	140	116	140	116
$^1/_4$	98	87		166	144		255	
$^5/_{16}$	199	178	275	352	306		454	
$^3/_8$	356	319	437	544	458		764	
$^7/_{16}$	565	501	622	925	774		1,152	
$^1/_2$	864	771	1,081	1,267	1,065	888	1,705	1,421
$^9/_{16}$	1,234	1,124		1,671	1,363	1,136	2,382	1,984
$^5/_8$	1,688	1,588		2,172	1,714	1,429	3,194	2,661
$^3/_4$	3,015	2,730		3,771	3,008	2,506	5,452	4,544
$^5/_8$	4,768	4,371		6,094	4,841	4,034	8,361	6,967
1	7,276	6,554		9,122	7,234	6,028	12,330	10,270
$1^1/_8$	10,325	9,295		13,450	10,480	8,735	17,260	14,390
$1^1/_4$	14,202	13,086		18,030	14,470	12,060	23,760	19,800
$1^1/_2$	25,102	22,686		32,240	25,050	20,870	39,530	32,940
$1^3/_4$		35,828		51,660	39,820	33,180	62,900	52,420
2		53,961		77,160	59,480	49,570	91,210	76,010
$2^1/_4$		78,762		110,800	85,660	71,380	129,800	108,100
$2^1/_2$		107,589		152,100	117,600	97,960	176,800	147,300
$2^3/_4$		145,761		204,200	164,000	131,700	233,700	194,800
3		192,514		265,100	203,300	169,400	306,900	255,700

[1] Plotted in Fig. 4.

The author believes that the foregoing facts and the following data justify considering the nut from the sole standpoint of transmitting the torque of the wrench. In doing so it is found that regular nuts proposed by the A.E.S. Committee are large enough under all ordinary conditions, but they do not have a sufficient factor of safety to ignore the checking of exceptional cases.

It will be noted that the first two columns of Table 4 give the wrench torque in inch-pounds that will load a bolt of given diameter to 60,000 lb. per sq. in. This will be the first matter for the engineer to consider, and the torque can be modified to the value necessary, as it is proportional to the load. The other columns of Table 4 give the torque-transmitting capacity of various nuts with the worst condition of fit between the nut and wrench allowed in the committee's report, and with similar tolerances on United States Standard nuts.

[3] This report on Tentative American Standards for Wrench Head Bolts and Nuts will be available in pamphlet form about November 1, 1926. An abstract of this report appeared in the May issue of MECHANICAL ENGINEERING.

Some of the above information is shown in Fig. 4. It will be noted on this chart that the scale at the right is in inch-pounds and varies as the cube of the diameter.

Since the torque requirements for bolts vary as the cube of the diameter (if nuts, bolts, and threads are designed directly in proportion to the diameter), they tend to lie in a straight line on this chart.

For comparison, the torque required to load bolts to 60,000 lb. per sq. in. stress is shown by a full line. This line would be straight if the pitch were in all cases exactly proportional to the diameter, so that it is in fact nearly a straight line.

The U. S. Standard nut width is directly proportional to the diameter plus a constant, so that the values of the torque which will be transmitted by these nuts lie in a straight line. The regular nuts $^5/_8$ in. and larger have a dimension directly proportional to the diameter of the bolt, so that their line of torque-transmitting ability is straight and inclined at a slight angle to the U. S. Standard line.

It will be noted that with the worst wrench fit allowed and the low Brinell which frequently occurs in hot-pressed nuts, hexagon nuts of the regular dimension do not have sufficient torque-transmitting ability to load a coarse-thread bolt to 60,000 lb. per sq. in. without rounding the corners by the wrench, while nuts of the same dimension with finished tolerances and cold-pressed material will have substantially the same torque-transmitting ability as U. S. Standard hot-pressed rough hexagon nuts.

Since the torque-transmitting ability of a nut depends upon its fit in the wrench and hardness as well as size, it will be seen that the regular cold-pressed hexagon nut will be satisfactory wherever the U. S. Standard hot-pressed rough hexagon nut has been satisfactory. In fact, it will probably be considerably better, as in these curves it is assumed that the U. S. Standard nuts have been made with the same tolerance that is now required for rough nuts in Table 4 and used with wrenches made within the same tolerance. As a matter of fact, the tolerance on bolts, nuts, and wrenches has been very irregular, and it is probable that those in service have frequently been much more loosely fitted than the tables would indicate. It has also been true that iron has been used for these nuts so that it is probable that many U. S. Standard nuts have been used in actual practice whose torque-transmitting ability has been considerably below that indicated by the broken line in Fig. 4.

All of these figures indicate that when nuts are to be selected the hardness and the workmanship on the nut and wrench as indicated by the tolerance must be considered as well as the outside dimensions of the nut, and that it is easily possible to use lighter nuts which will be fully as satisfactory as the heavier ones now in common use.

The way in which the workmanship (as indicated by the dimensional tolerance) and the material (as indicated by the hardness) affect the torque-transmitting ability of nuts and, therefore, their satisfactory use will now be shown in more detail, and a careful study of these data should make it possible to purchase nuts of such material and workmanship that the lighter ones will be even more satisfactory and considerably less expensive that the U. S. Standard nuts formerly used.

According to the tables of the report of the Sectional Committee on Bolt, Nut, and Rivet Proportions, the value of the ratio of W (maximum wrench opening) to N (minimum nut width) for rough nuts is very close to 1.05 in the worst case. For finished nuts it approximates 1.04 on the three small sizes, and then gradually decreases to 1.02 for the 3-in. nut.

The torque-transmitting abilities of nuts are dependent upon the hardness of the stock, and so will vary as the composition of the material varies. It would seem that there is an opportunity for the designer to use the most economical nut by considering whether the nuts are to be used with open-end wrenches, and whether he can depend upon the wrenches and nuts having the tolerance given. Careful selection would seem to make the use of United States Standard nuts unnecessary except in extreme cases.

While the regular and the light nuts would not be satisfactory with the extremely wide tolerance on nuts and wrenches which has been customary, it would appear that the regular nuts with the Committee's tolerance should be satisfactory for substantially all purposes, even when used with open-end wrenches, and that the light nuts should also be satisfactory in many such cases and in all cases when used with socket wrenches. The tolerance given in the Committee's tables should not be difficult for either the nut manufacturer or the wrench manufacturer to manufacture to. The data given will make it possible to select the most economical nut in any case.

The fact that the torque-transmitting ability is the limiting factor rather than the strength of the nut or the strength of the thread, explains something which has baffled many who have investigated this matter of standardizing nut sizes.

Some have asked why the automobile standard nut cannot be used in all cases. It is believed that the tables answer that question. They also explain why Europeans use much heavier nuts than Americans. There is no question of torque-transmitting ability on a square nut, and 95 per cent of the nuts furnished with bolts in America are square, while in Europe hexagon nuts are used almost entirely, and many of them are very rough. The lighter square nut gives universal satisfaction. The lighter hexagon nut is satisfactory only with close tolerances and good material for both the nuts and wrenches, or when used with socket wrenches. Both of these conditions are usually found in the automobile industry.

In line with this we find that, as usual, practice coincides with theory. It is the experience of the company with which the author is connected that 75 per cent of the hexagon nuts which it makes are of the full U. S. Standard dimensions, while only 10 per cent of the square nuts are. The data in the tables indicate that these nuts with the tolerance recommended will be as satisfactory as the U. S. Standard nuts with the wider limits now used.

It is believed that the wrench and nut tolerances given are as close as can be enforced with the present-day practice of manufacturers. Very little attention has been given in the past to tolerance either on the nuts or on the wrenches.

It will be noted that hot-forged U. S. Standard nuts made of material testing about 116 Brinell give almost the same torque as the Committee's regular nuts with closer tolerances and harder material. It would appear from the tables that the light series will be satisfactory where socket wrenches are usually used and nuts are of the better grades, and that the regular series of nuts should be satisfactory for any purpose where socket wrenches are used, and with open-end wrenches for any condition where it is economical to use the better-finished regular nut rather than the rough-forged U. S. Standard nut. In fact, about the only condition that makes it necessary to use the U. S. Standard nut is when rough nuts must be used and the bolt is to be loaded with the wrench, especially when the bolt is made of material better than regular bolt steel.

It may be assumed that square nuts of the sizes mentioned in the report of the Sectional Committee will be entirely satisfactory. There seems to be no reason for using anything large than the regular square nut under any ordinary conditions, as 90 per cent of the square nuts now in use are of about these sizes.

In the tables the mathematical information necessary for the selection of proper nuts is given without showing the rational method used in its development. The following paragraphs consider this in detail.

The ability of a hexagon nut to transmit torque depends upon its width and thickness, material (which may vary in hardness from 116 to 200 Brinell in ordinary commercial nuts), and upon its fit in an open-end wrench, which will be from 2 per cent to 6 per cent larger than the nut in the extreme case.

The ability to transmit torque will be closely proportional to the Brinell number. The variation in wrench fit mentioned above gives a torque-transmitting ability of about two-thirds more with 6 per cent clearance than with 2 per cent. Rounded and gouged corners make surprisingly little difference in the torque available. The amount of rounding of corners allowed by the Committee's tables does not decrease the ability of the nuts to transmit torque.

Tests made on readily accessible nuts showed the hardness to be fairly constant for each class, but some makers use a steel of higher carbon content for hot-pressed nuts, thereby increasing the hardness, and some nuts are made on a screw machine from cold-drawn bessemer stock, and this material may test 140 to 200 Brinell.

Fig. 5 is a diagram of a wrench turning a nut. The torque transmitted to the nut by the wrench is equal to the product of the force acting on the surface AB times twice the radius (equal to the

line EH), through which it acts around the center of the nut. The force is equal to the compressive strength per square inch under the conditions times the area of the surface. The area is equal to the product of the length of the line AB and the thickness of the nut. The radius EH, through which the force acts, is equal to twice the length of the line RA plus the length of AB, divided by two.

For convenience in calculating, the factors Q and H are introduced:

$$Q = \frac{2(2RA + AB)}{2} AB \qquad H = AB$$

Q is actually the torque which would be transmitted by a nut of unit thickness, unit width, and a compressive strength of a unit amount per unit of area under the conditions. In these calculations this would mean that Q is the torque transmitted by a hexagon nut 1 in. wide, 1 in. thick, and of a compressive strength of 1 lb. per sq. in. Referring again to Fig. 5, which is assumed to be a drawing of a nut of unit dimensions, it is clear that if we know the amount of torque which a nut of unit dimensions will transmit, the torque-transmitting ability of any other nut (with a given ratio of W to N)

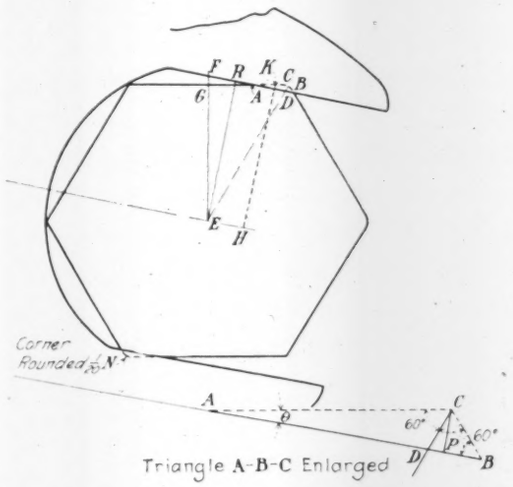

FIG. 5 DIAGRAM ILLUSTRATING DEVELOPMENT OF RESISTANCE TO TORQUE

will vary from it as the square of its width, directly as its thickness, and directly as its compressive strength per unit area; therefore the torque in inch-pounds becomes

$$Pl = QN^2T_hB$$

where N = width of nut across flats, in.
T_h = thickness of nut, in.
B = compressive strength of material, lb. per sq. in. (see Table 6)
H = distortion of nut by wrench
P = pull on wrench, lb.
l = effective length of wrench, in.

The relation between Q and H can be determined mathematically, but the author has been unable to develop any formula with less than 18 terms. The relation was found by laying out a 1-in. hexagon magnified on a scale of 50 to 1, and graphically determining the amount of distortion H of the nut as the wrench was revolved around the hexagon. By measuring the distortion H (a measure of the force applied) and the distance through which it worked, the torque constant Q was determined.

Fig. 6 shows the relative value of Q and H for certain values of the ratio of W to N as the wrench is turned around the nut a certain number of degrees. The solid lines represent the value of Q and the dotted lines the values of H.

This chart is interesting in showing the number of degrees which a wrench bites into the nut in creating a certain torque. This number of degrees is a loss when considering the clear angle through which a wrench must be swung in order to rotate a nut. The angle of play is not a loss.

Fig. 7 is the basis for all of the tables of torque-transmitting ability by wrenches and nuts. It shows the relation between Q and H for various values of the ratio $W:N$. It is interesting in showing how the torque Q transmitted by the wrench increases at first almost directly as the distortion H, but soon slows up and finally decreases as the wrench rounds the corner. The curves start about

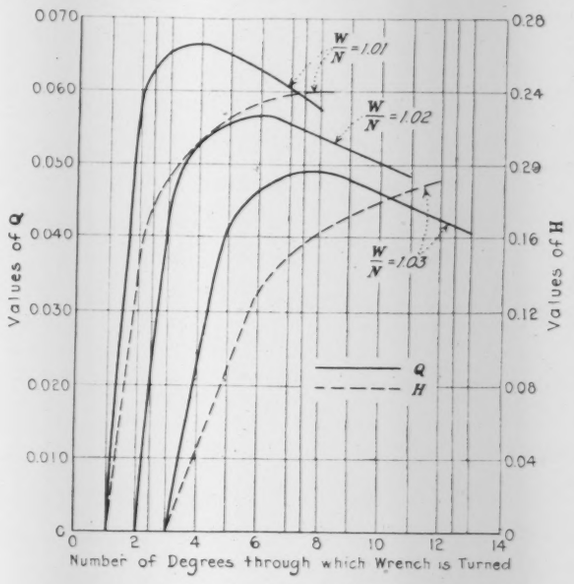

FIG. 6 RELATIVE VALUES OF Q AND H FOR CERTAIN VALUES OF W/N AS WRENCH IS TURNED ROUND THE NUT A CERTAIN NUMBER OF DEGREES

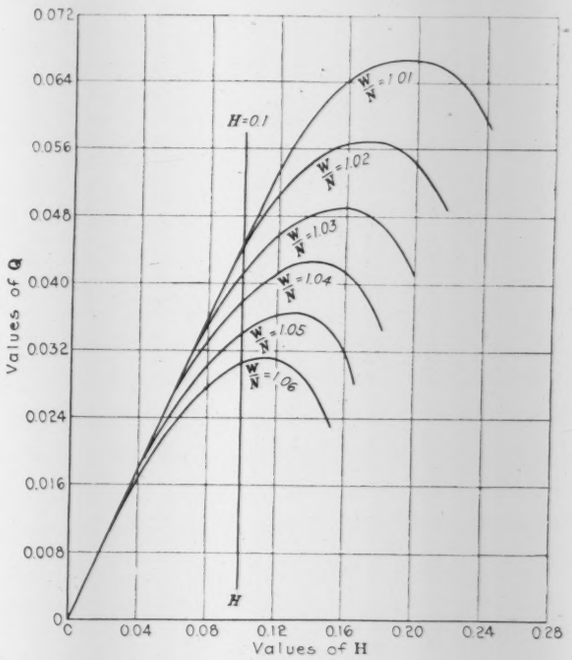

FIG. 7 CURVES WHICH ARE THE BASIS FOR TABLES OF TORQUE-TRANSMITTING ABILITY BY WRENCHES AND NUTS

the same for all of the wrench clearances, but as the ratio of the wrench width W to the nut width N increases, the torque falls off rapidly compared to the distortion produced.

The value of H shown by the vertical line represents the point at which the wrench will have flattened a hexagon nut at the corner for a space 0.1 of the width of the nut across the flats (0.1 in. on a nut 1 in. across flats). This value is used in the tables as a maximum since it seems about the maximum allowable distortion.

It will be noted that the values of B represent the compressive strength of the material of the hardness given under the conditions of the test. This figure is considerably higher than the compressive strength of the steel under ordinary conditions, but apparently

where the force is introduced at a slight angle to the surface, the material does not flow as readily.

Values of B were determined experimentally by compression tests in a testing machine in the jig shown in Fig. 8, and checked by turning a wrench around the corners of nuts of known hardness. In the table it is assumed that values of B are proportional to Brinell numbers. Very few tests were made on this point, but they seem to check fairly well, and it is reasonable to assume that the flattening by a wrench is very similar to that produced in a Brinell testing machine. Values of B are shown in Table 3.

It can be proved graphically that rounding the corners of the nut to the extent of a radius $1/20$ its width has no effect on the torque.

In the tables of the Sub-Committee the minimum value given for the distance across the corners of the nut is based upon such a rounding of the corners giving a minimum width of 1.14 times the minimum long diameter of the nut. Any larger nut will produce more torque, so that there is no objection to more rounding on the corners of larger nuts; in fact, there are many considerations in the design of socket wrenches, and even of open-end wrenches, which

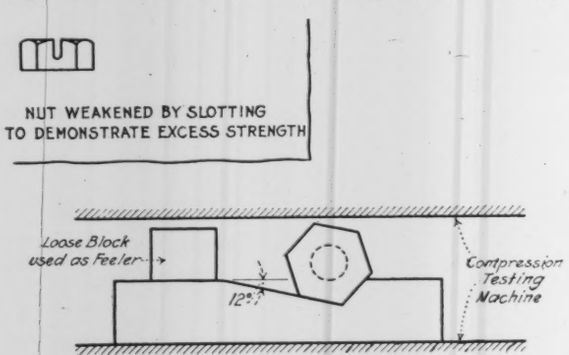

FIG. 8 JIG FOR OBTAINING EXPERIMENTALLY VALUES OF B (COMPRESSIVE STRENGTH OF MATERIAL IN POUNDS)

would make it desirable for nuts to be made regularly with considerably more rounding.

Testing for torque-transmitting ability it was found that it took considerably more torque to force a wrench around a bar than it did to force it around a nut with a hole in it, but it was found that a nut on a bolt acts like a solid bar.

This theoretical work was checked by measuring the torque necessary to turn a wrench around the corners of nuts on the machine at the plant of The Lamson & Sessions Company.

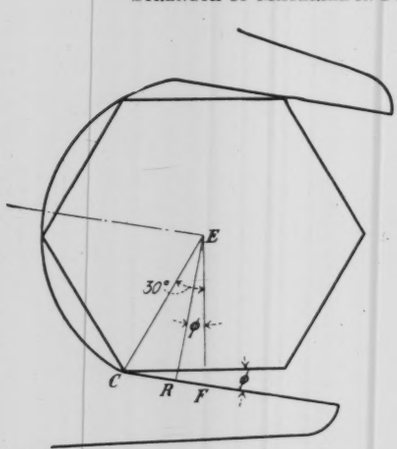

FIG. 9 FREE ANGLE THROUGH WHICH WRENCH CAN TURN FROM CENTER POSITION BEFORE CONTACTING WITH NUT

Free Angle. The free angle is the angle through which the wrench can turn from a center position before contacting with the nut. Note that it can turn an equal amount in the opposite direction, making a total play of twice what is here called the free angle, shown in Fig. 9 which is used to develop the free-angle formula.

$$\cos(30^\circ - \phi) = \frac{ER}{EC}$$

$$ER = \frac{W}{2}$$

$$EC = \frac{1.1547\,N}{2}$$

$$\therefore \cos(30^\circ - \phi) = \frac{W}{1.1547\,N}$$

Values of ϕ for various values of the ratio of W to N can be found in Table 5.

The free angle does not affect the space necessary to swing the wrench. A 15-deg. wrench can turn a hexagon nut with 30 deg. clear swing and a $22\frac{1}{2}$-deg. wrench with a 45-deg. swing will turn a square nut. Any flattening of the corner by the torque of the wrench will increase the clear swing necessary.

TABLE 5 VALUES OF FREE ANGLE Q AND H FOR VARIOUS VALUES OF RATIO $W:N$

$\frac{W}{N}$	Free angle ϕ	Q at $H = 0.10$ (Fig. 7)	Q Max.	H at max. torque	H at max. value
1.01	1°	0.0450	0.0668	0.208	0.240
1.02	2° 3′	0.0440	0.0568	0.178	0.220
1.03	3° 7′	0.0405	0.0492	0.172	0.198
1.04	4°14′	0.0370	0.0424	0.142	0.181
1.05	5°25′	0.0335	0.0366	0.120	0.165
1.06	6°22′	0.0300	0.0314	0.110	0.150
1.07	7°55′	....	0.0262	0.106	0.136
1.08	9° 7′	....	0.0212	0.088	0.120
1.09	10°35′	....	0.0164	0.076	0.104
1.10	12°17′	....	0.0126	0.054	0.088
1.11	14° 0′	....	0.0082	0.044	0.073
1.12	15°55′	....	0.0065	0.034	0.058
1.347	30° 0′	....	0.0000	...	...

TABLE 6 VALUES OF B FOR VARIOUS BRINELL HARDNESS NUMBERS

Brinell hardness number of nut	Value of B	Brinell hardness number of nut	Value of B
120	102,000	170	146,000
130	111,000	180	154,000
140	120,000	190	163,000
150	128,000	200	171,000
160	137,000		

TABLE 7 DATA FOR USE IN WRENCH DESIGN

Nom. diam. of bolt, in. D	Overall length of wrench, in. $16D$	Effective length of wrench in. $16D-1$ $16D-2$	Torque required to load bolt to 60,000 lb. per sq. in. (Coarse thd.), in.lb.	Pull on standard wrench lb.	Length of track bolt wrench, in.	Pull on track-bolt-wrench, lb.	Pounds force opening wrench jaw, lb.
1/4	4	3	87	29	..	..	12,800
5/16	5	4	178	44	..	..	17,400
3/8	6	5	319	64	..	..	20,100
7/16	7	6	501	84	..	..	19,500
1/2	8	7	771	110	..	..	41,000
9/16	9	8	1,124	141	..	..	46,300
5/8	10	9	1,588	176	..	..	52,500
3/4	12	10	2,730	273	..	..	63,200
7/8	14	12	4,371	364	40	109	74,000
1	16	14	6,551	468	42	156	84,700
1 1/8	18	16	9,295	581	44	211	97,000
1 1/4	20	18	13,086	727	48	273	106,200
1 1/2	24	22	22,686	1,031	60	378	127,700
1 3/4	28	26	35,828	1,378	..	..	149,100
2	32	30	53,961	1,799	..	..	170,600
2 1/4	36	34	78,762	2,317	..	..	192,100
2 1/2	40	38	107,589	2,831	..	..	213,600
2 3/4	46	44	145,761	3,312	..	..	333,000
3	48	46	192,514	4,185	..	..	256,500

THE DESIGN AND USE OF WRENCHES

It is very important that wrenches be properly selected. For reasons given previously, it is recommended that bolts, nuts, and wrenches be so designed that a 50-lb. pull on the wrench will develop the necessary grip in the bolt while a 200-lb. pull on the wrench will be necessary to injure the bolt, because men tightening bolts are apt to pull anywhere from 50 lb. to 200 lb. This means a factor of safety of four in the bolt, which does not seem excessive. These values may be reduced by half for $1/4$-in. bolts and smaller, and to a less extent for bolts $1/2$-in. and smaller.

Table 7 gives the nominal bolt diameter and the average overall length of standard wrenches in the first two columns. Column 3 assumes the effective length of the wrench as from one to two inches less than the overall length, according to the size. The other five columns are all figures in pounds or inch-pounds required to load bolts to 60,000 lb. per sq. in. Other values will be proportional, as the torque is directly proportional to the load.

It will be noted that the breaking pull on the standard wrench is so small for bolt $5/8$ in. or under that there is great danger of breaking the bolt in an attempt to tighten it, while it is hardly conceivable that a sufficiently long wrench can be provided to tighten bolts $1\frac{1}{4}$ in. and longer to anything like their full capacity.

The wrench designer is faced with two problems: one, the provision of a handle suitable as to length and strength to take care

of the torque involved, and the other the stresses tending to open the jaw of an open-end wrench.

Whenever a wrench is provided for a certain job its length should be most carefully selected with regard to the figures given in columns 4 and 5 of Table 7.

Tests on wrenches on the market have shown that many are none too well designed as to stiffness of the jaws. Fifty different types of wrenches all with a 1-in. opening were purchased from various jobbers and other sources, and of them only a few were as stiff as they should be, while the opening in the end of the wrench varied all the way from 0.995 in. to 1.020 in. Holding wrenches and nuts to the tolerance given earlier in this article will more or less reduce the strain tending to open the jaw when the nut is tightened. Fig. 10 indicates a shape of jaw opening of greater strength than the usual design when used for hexagon nuts. The stresses tending to open the wrench when using a square nut are much lower, so that such a wrench will be amply safe with square nuts if designed for hexagon nuts.

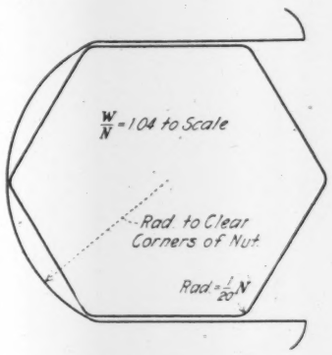

FIG. 10 SHAPE OF WRENCH-JAW OPENING OF GREATER STRENGTH THAN USUAL DESIGN WHEN USED FOR HEXAGON NUTS

The relation of the load in pounds tending to open the wrench to the torque is shown in columns 4 and 8 of Table 7. It is to be noted that the designer should not use these figures directly in designing the jaws of his wrench for stiffness. He should consider first what torque will break the bolt, and then the length of the wrench handle and the pull which is likely to be applied to it. If the wrench will be used directly, 200 lb. may be taken as a maximum of such force, but if it is of such a shape that a pipe can be applied, the designer will have to figure on a corresponding increase in torque.

When the larger bolts are considered it will be found that it is improbable that any such torque as that given in column 4 will be applied, and the opening force will then be reduced in direct proportion.

It will be noted that all of the figures in the previous sections dealing with the torque-transmitting abilities of nuts and the design of wrenches are based upon wrenches and nuts made with the loosest condition of fit allowed by the Sectional Committee's Report. If wrenches are made with a greater tolerance or if they are made so light that they will spring, the force tending to open the jaws is very much increased and the liability of severe damage to the nut is much greater. Buyers of wrenches and designers of wrenches should consider these figures and check them carefully, as railroad men and others will testify that there is very frequent trouble with wrenches which turn around the corners of hexagon nuts.

In order to be entirely safe it will be necessary for the wrench designer to assume that the force computed as above will be applied at the extreme point of the jaw.

The suitable selection of nuts, bolts, and wrenches adapted to each other and to the use to which they will be applied is very necessary, and it is hoped that this article will make possible such selection. While the mathematics involved have been worked out in detail, it is unnecessary to use them except under exceptional circumstances. The tables are intended to give the necessary information in such a way that it can be readily found and applied. It should be remembered that they assume a lubricated bolt with a coefficient of friction of approximately 0.15, stressed below the point where it commences to stretch, and used with nuts and wrenches made with the tolerance given in the tables in the Sectional Committee's report.

In concluding, the author would like to lay particular stress on the fact that for bolts smaller than $^5/_8$ in. the principal danger is from breakage. For bolts from $^5/_8$ in. to 1 in. perhaps the principal danger is that the corners of the nuts will be rounded. On bolts over 1 in. in diameter it is unlikely that sufficient torque will be used to tighten the bolt efficiently. If the present article assists bolt and nut designers and users in avoiding those dangers, it will have served its purpose.

Maintaining Production

IN order to maintain a successful production schedule, there must be coöperation between the production department and the departmental foreman. The foreman must report progress from time to time, and notify the production department immediately, if there is a hold-up, so that the necessary steps may be taken to clear obstructions and devise some means whereby the schedule may be maintained. This is especially true in the furnishing of tools, which is generally the reason why many schedules fail.

Centralizing the responsibility for all production in the production department gives a direct control of the output, as the personnel of the production department acts as the "go-between" of the different departments. It often happens that one foreman feels he does not want to take what he thinks are orders from another foreman, but if these orders came direct from the production department, there is no logical reason why action should not be taken, as all instructions regarding production emanate from the same source.

In the operation of any factory, there is necessarily a certain amount of wastage and spoilage. Wastage is material that is unavoidably destroyed in the process of manufacturing, but steps must be taken to keep this wastage down to a minimum. Scrap must be watched closely, and the pieces scrapped must be replaced. Just as soon as the inspection department scraps anything, the production department must be notified, so that replacements can be made immediately.

The unit cost of any product decreases as the quantity of production goes up; but over-speeding, especially by men on piece-work, must be watched closely, as in such cases quality is often forgotten in thinking about quantity. The workers should be given to understand that all work rejected must be replaced; only work that will pass inspection will be paid for.

The running time of equipment should be watched closely, and idle machinery and equipment reported to the production department. In one factory where there was a great deal of small lathe work, one department was so overcrowded that it was impossible for it to meet the schedule without working a great deal of overtime. This condition had been going on for several weeks before it was discovered that another department had idle lathes that could do the work. Transferring some of the work from one department to the other enabled both departments to keep busy, and the schedule was maintained without the use of over-time.

It should be possible to obtain the necessary production during the regular day's run, and not make a habit of working long hours. A man can stand the strain of extra hours for a limited time, but after that his productive effort decreases. Should it be found necessary to work longer hours in order to maintain the production schedule it is best to put on a double shift, even when this may call for a night shift on one or two operations only. Over-time can be excused for a short time, but over a long period it becomes quite costly. It costs more per hour in wages, the effort per hour is reduced, and the possibility of accidents greatly increased.

The workman must be suited to the work he is asked to perform if the best results are to be obtained. On a certain job, where it was necessary to wheel a great deal of concrete, it was noticed that the tall men spilled a part of the load before reaching its destination, due to the angle at which they had to hold the wheelbarrow when walking. When shorter men were put on the job, practically no concrete was spilled during the wheeling process. This is of importance in all lines of work. If a man is too short for the bench on which he is working, give him a platform to stand on, or a lower bench; in any case, if the best results are sought, the workman must feel at ease. (*Machinery* (N. Y.), vol. 32, no. 12, Aug., 1926, p. 969.)

Industrial Boiler Efficiencies

TWO papers dealing with industrial-plant problems were presented at the Providence Meeting of the A.S.M.E., Providence, R. I., during the first week of May. In the first of these, on industrial-boiler efficiencies, the author, Samuel D. Fitzsimmons[1] gave results that had been obtained with two 300-hp. water-tube boilers installed in the plant of the Brown & Sharpe Manufacturing Company in that city. The purport of the author was to indicate the desirability of low furnace-draft velocities and of a more effective use of radiant heat, and he assumed as a premise that more width in the tube sections of water-tube boilers was desirable. Much that was offered in the paper was in the form of conjecture. Mr. Fitzsimmons' paper was published in the May issue of MECHANICAL ENGINEERING, p. 413. A summary of the voluminous discussion it drew forth immediately follows.

Geo. H. Barrus[2] submitted a written discussion in which he said that the paper furnished a telling example of the excellent evaporative results that could be obtained by giving suitable attention to the correct principles of boiler engineering and operation. First, there had been efficient automatic stoking, which undoubtedly had secured a high degree of uniformity in the feeding, distribution, and combustion of the coal, with a small amount of waste carbon at the dump grate. Second, there had been a low air blast beneath the fuel beds, while a minimum force of draft had been required at the regulating dampers and within the furnaces. Third, the baffling had been arranged to give a gradually lessened area of gas passage extending from furnace to flue, whereby short circuits had been obviated and the entire heating surface made effective. Fourth, the boilers had not been forced beyond an economical rate of production. These features had all combined to secure a high furnace temperature, a low flue temperature, and a good showing in the gas analysis, which were three essential elements required for high evaporative efficiency.

W. T. Hatch[3] wrote that the emphasis the author placed on the advantageous use of radiant heat was very timely, and the remarkable efficiency secured was no doubt due in large measure to the judicious arrangement of the heating surface directly over and virtually parallel to the fuel bed. Another important principle that this installation illustrated was that highly efficient combustion could be secured with very low blast pressure.

With popular interest running so strongly in favor of the use of pulverized fuel, it was fortunate for the engineering profession to have definite proof that equally high efficiencies could be secured with solid fuel on underfeed stokers, and probably at substantially lower cost both for installation and maintenance.

Furthermore, the radiant heat could be more directly utilized with the stoker than with pulverized fuel, and it was possible that a reduction of the distance between the stoker and the boiler tubes might make a still better showing for the stoker-fired boiler.

E. N. Trump[4] wrote that after comparing the author's tests with those on a B. & W. boiler reported by B. N. Bump,[5] he found the difference between the two was largely in the percentage of tube surface exposed to radiation. He believed, therefore, that the author was right in his conjecture that improvement in boilers was to be found in an increase in the amount of surface exposed to direct radiation from the fire.

However, it seemed to Mr. Trump that there was no reason why the stoker furnace should give any more effective radiant heat than the pulverized-fuel furnace, if combustion was perfect in the latter. The boiler of the future would evidently consist of heating surface surrounding the furnace on all sides, with an envelope of air which would receive the heat of outside radiation on its way to the furnace. The increase of boiler pressures which raised the temperature of the water where the gas made its exit, increased the loss from the furnace gases and made the air heater or economizers increasingly necessary. If two pressures were used it might, in many cases, be possible to introduce the heating surface for the high pressure so that it would be exposed to direct radiation from the fire and the highest temperature, and then pass the gas through the low-pressure boiler, so that by pumping the water in the opposite direction, first into the low-pressure boiler and then into the high-pressure boiler, the heating might be progressive and the gas leave the boiler at the temperature of the low-pressure water.

James J. Tyrrell,[6] in a written discussion, said that all too often boilers were purchased on the basis of total heating surface in the unit, with little, if any, consideration of the distribution of heating surface with relation to that which was exposed to the fire and which absorbed heat by radiation and that which must depend solely on convection for its absorption power.

The author noted particularly the comparatively low settings of these boilers. It should be borne in mind, however, that furnace height was a function of furnace volume, and it happened that in this particular boiler ample furnace volume for the fuel burned was provided by this height of setting.

In discussing requirements of stoker furnaces as compared with those of pulverized-coal furnaces, the author had remarked that some one had advanced the idea that equal volumes were required with both fuels. It seemed to Mr. Tyrrell that this was obviously not so as, since pulverized coal was burned in suspension and while the coal was in motion, larger furnace volumes must be provided for this kind of fuel in order to obtain complete combustion in the furnace because of the time element involved in the velocity of the fuel in the furnace. Of late, improvements in burning arrangements had been made which permitted the use of greatly reduced furnace volumes, and it was his opinion that while furnace volumes required for pulverized coal would be somewhat in excess of those for stokers, they would be somewhat less than those used in the past and would in general conform to the requirements of fuel oil.

The author referred to the destructive effect of radiated heat on uncooled furnace walls. It would seem that in many cases furnace designers had not considered refractory linings of furnaces in any other way than as structures which would withstand some degree of heat, whereas properly placed firebrick linings would reflect heat to the boiler after becoming incandescent, and, by sloping and otherwise arranging walls with this in mind, longer life could be secured from the refractories and the performance of the boiler bettered without sacrificing useful combustion volume to any great degree.

In Mr. Tyrrell's opinion, the really important thing brought out by the author was indicated by his statement that much set forth was conjecture. There was little information of value which would indicate the maximum absorption power of a square foot of clean heating surface when exposed to radiant heat. There seemed to be no definite knowledge relating to the permissible draft loss through boilers. There seemed to be no consistency to the figures available pertaining to the furnace-volume requirements of different kinds of fuels or different firing methods

This lack of information available to the industrial-plant engineer was a serious obstacle to securing the best economies in the boiler plant. Knowledge of combustion, suitable apparatus for insuring proper furnace conditions, and intelligent checking would aid in operating a plant up to the point of its individual high efficiency, but it would seem that it was at least of equal importance to have data available which would make possible an intelligent choice from the equipment offered.

James W. Armour[7] wrote that the paper was very timely in view of the great interest which was now being shown in the application of pulverized fuel to steam boilers. Pulverized-coal development, as applied to steam boilers, had been very rapid during the past few years, and still further rapid development could be looked for. Stoker development had now extended over a period of about forty years so that, naturally, developments and refinements of the past few years had not been as marked, and therefore had not aroused as great interest.

It was interesting to see what Brown & Sharpe Manufacturing

[1] Plant Engineer, Brown & Sharpe Mfg. Co. Mem. A.S.M.E.
[2] Consulting Steam Engineer, Boston, Mass. Mem. A.S.M.E.
[3] Providence, R. I.
[4] Mechanical Engineer, Syracuse, N. Y. Mem. A.S.M.E.
[5] Trans. A.S.M.E., vol. 34 (1912), p. 687.
[6] Pawtucket, R. I.
[7] Engineering Manager, Riley Stoker Co., Worcester, Mass. Assoc. Mem. A.S.M.E.

Company had done in the way of obtaining high boiler efficiencies without going to additional expense for those refinements which were generally considered necessary for such high efficiencies. The results obtained at this plant were furthermore interesting in view of the tendency, during the past few years, toward higher boiler settings and larger combustion spaces. By keeping the boiler setting comparatively low, they had obtained low maintenance of brickwork and increased effectiveness of the boiler as a radiant-heat absorber, and had not had to spend money for raising the boilers or enlarging the building.

The author pointed out that the present design of his setting was very efficient up to ratings of approximately 200 per cent, but beyond this point the efficiency dropped off rapidly, due to increased combustible in the ash. Here Mr. Armour felt that efficiency could be kept up to good central-station practice by increased investment. There were two things which could be done. First, a longer stoker would give a longer fuel travel and therefore a longer time in which to burn out carbon before the material reached the dumping point. Second, by the use of clinker grinders, the combustible in the ash would be reduced, so that the loss from this source should not be proportionately any greater than at lower ratings, but it was then a question whether this increased return in efficiency was going to pay as big dividends as an equal investment in other equipment about the plant.

Mr. Armour agreed with the author that as high efficiencies could be obtained with stokers as with pulverized fuel, provided conditions were equal. However, he believed that high efficiencies were more easily obtainable and furnace conditions more easily controlled with pulverized fuel than with stokers, and that pulverized fuel would permit the use of more varied types of fuel without sacrificing efficiency. Comparatively large combustion chambers were necessary at the present time with pulverized fuel, and this of course increased the investment. What the future would bring could only be predicted, but it was quite possible that development in pulverized fuel would evolve a type of burner and improvements in boilers so that smaller combustion chambers could be effectively used and the cost of such an installation could be brought down equal to or lower than the cost of a similar stoker installation.

George W. Bach[8] wrote that while the performance of the Brown & Sharpe boilers was remarkable, there were other Union water-tube boilers operating equally as well in other plants, one of these being located at the new Riverside Pumping Station, Milwaukee, Wis.

A comparison of the boilers at Milwaukee with the Brown & Sharpe boilers was quite consistent, as both installations had the same make of stokers of the same size, and the test results at Milwaukee had averaged identically the same CO_2 condition, making this an unusually fair opportunity for comparison. The boilers at Milwaukee were somewhat larger, being 412 rated horsepower, and these boilers showed an exit-gas temperature only 27 deg. higher than the saturated-steam temperature, which was 390.2 deg. fahr. at 206 lb. gage operating pressure at 110 per cent of rating.

Observing the heat balance of the Brown & Sharpe test, it would be noted that there were but two losses attributed in any way to the boiler. These were (1) loss due to dry chimney gases; (2) loss due to unconsumed hydrogen and hydrocarbons, radiation and unaccounted for, forming respectively 10.1 and 1.65 per cent, or a total of 11.75 per cent of the heat given the stoker. Of this amount the item of unconsumed hydrogen and hydrocarbons was not chargeable to the boiler. The remainder of the losses, or those which might be charged against the stoker, were 5.08 per cent. Then the efficiency of the stoker would be 94.92 per cent and the efficiency of the boiler and setting alone, 87.62 per cent.

Analyzing the Milwaukee Water Works test in the same way, the losses due to dry chimney gas were found to be 9.69 per cent, and the unaccounted losses, 4.51 per cent, or a total of 14.20 per cent, leaving the percentage chargeable to stoker 8.29 per cent. The efficiency of the stoker in this case was 91.71 per cent. The efficiency of the boiler was 84.34 per cent.

In comparing these boiler efficiencies of 87.6 per cent and 84.3 per cent for the Brown & Sharpe and Milwaukee Water Works tests it should be borne in mind that the settings of the first had the furnace side walls insulated only up to the bottom row of tubes, while the Milwaukee boiler had a steel casing with 2 in. of insulating material between it and the brick, all over. The Brown & Sharpe boilers had been operated at 127 lb. pressure and the Milwaukee at 206 lb. gage.

The Brown & Sharpe boiler had had 10 ft. 6 in. length of tubes, 17 in number, exposed to radiant heat; the Milwaukee boiler 9 ft. 8 in. length, 17 in number. The builder's ratings had been respectively 308 and 412. The heights under the front header were 9 ft. 6 in. and 12 ft. 0 in.; widths of furnace, 7 ft. $7\frac{1}{2}$ in. and 8 ft. 0 in. The Milwaukee boiler had a superheater, while the Brown & Sharpe did not.

In general Mr. Bach believed that the difference in efficiency of 3.3 per cent between the Milwaukee boilers and the Brown & Sharpe boilers was due in great measure to the difference in temperature heads, caused by the different operating pressures in the two plants.

H. M. Burke[9] wrote that the paper presented a splendid example of the worth of a good mechanical stoker, for in considering the furnace volume, heating surface, circulation, and flue-gas temperature, and comparing with other makes of boilers of equal standing, very little if anything was found that had a tendency to bring the boiler efficiency much above ordinary good practice.

On the other hand, when the stoker and boiler were considered together, it was plainly seen that the furnace losses had been reduced. It would be interesting to have similar test results on the same boilers, hand-stoker fired.

Today hand stokers were very efficient in permitting proper mixture of fuel and air. At this rating and with this size of plant—a two-boiler plant—the labor cost would not change as about one ton of fuel per hour was used. The furnace volume in this case limited the economical high rating, and the saving in first investment and cost of repair parts were reasons given in favor of hand stokers.

In summing up these advantages and disadvantages, it would seem that in case of moderate-size installations, such as the two of this plant, great weight should be given the difference in investment between hand stokers and mechanical stokers. Also, the increased efficiency, if any, of the mechanical stoker over the hand stoker, properly operated, if the boilers were to be operated at such a modest overload, warranted serious consideration when small boilers were used. This would especially apply to plants operating a ten- or twelve-hour day.

Wm. G. Starkweather[10] contributed a written discussion in which he said that the furnaces had a volume of about $2\frac{1}{2}$ cu. ft. per rated horsepower, which had been considered rather unfavorable at first for attaining the very highest efficiency. Nevertheless this installation had set a high mark in heat utilization—83 per cent without superheater, economizer, etc.

Under initial conditions there had been some slag formation in the lower rows of tubes. While less air pressure had greatly reduced the slag-carrying capacity of the gases rising from the fuel bed into the tube body of these two boilers, the two new units now nearing completion had the lower row wider spaced to form a "slag screen," permitting more gradual reduction of gas temperature and opening up more tubes in the second and third rows, for better heat absorption, on a wider angle of the radiant-heat rays, then heretofore. There would probably be larger active heating surface in this installation than in usual designs.

Another leading feature of this boiler was the adequate supply of circulating water for all the tubes, particularly to the lower rows in the first pass where the major evaporation occurred. The rear connecting necks had a cross-section of 480 sq. in. each, and the entrances were rounded, thus avoiding contraction. The result was a solid stream of water pouring down into the rear header. This header was 10 in. wide, and distributed amply as demanded by the tubes, which of course required different quantities, depending on locations. At the front tube ends the emulsion of steam and water discharged into another wide header which delivered upward through a large unrestricted connecting neck with rounded

[8] Vice-President and General Manager, Union Iron Works, Erie, Pa. Mem. A.S.M.E.

[9] Plant Engineer, Mt. Hope Finishing Co., North Dighton, Mass. Mem. A.S.M.E.

[10] Starkweather & Broadhurst, Inc., Boston, Mass. Mem. A.S.M.E.

edges into the front end of the steam drum, having a surface of 123 sq. ft., for free release of the steam.

With $6 coal this installation was producing 1000 lb. of steam for less than 25 cents, which was a remarkable result for an industrial plant.

F. W. Dean[11] wrote that there was no statement in the paper of the method of determining the water used, but without doubt it had been metered. Somehow or other metered water usually gave high results.

Mr. Dean strongly disapproved of the author's remarks concerning radiant heat. Such statements were now frequently made and were causing an overvaluation of the importance of radiant heat. The fact of the matter was that radiant heat had no more value than any other heat. It was merely an accompanying feature of high temperature. If this temperature could exist without incandescence the result would be the same. A certain number of units of heat were given to the gas by the combustion of the fuel, and when some of them were absorbed by the surrounding heating surface, there were some left to be absorbed further on. A great quantity was absorbed above the grate simply because their temperature was very high and the head of heat was great. The same absorption of heat would be made by the tubes if there were not surface to receive it before the tubes were reached. Where did the mysterious heat in radiant gases come from, and how could there be any but B.t.u. from combustion? If there were heat that was anything but B.t.u. from combustion, it came very near violating the law of the conservation of energy. It made no difference whether a boiler had much or little surface exposed to radiant heat, its economy would be the same.

Some years ago, in Germany, two locomotives had been fitted with boilers without metal fireboxes but with brick fireboxes in the same location, and the results in service had not been noticeably different.

The author said that high furnace temperature was necessary for economy. Mr. Dean doubted this very much. His belief was that economy came from burning the carbon to CO_2 with minimum air, perfect circulation of the resulting gases around or through the heating surfaces, and the utilization of all the heating surface.

The author also said that the circulation in this boiler was very rapid and that this was a requisite to economy. The design of this boiler, however, was such that there was considerable resistance to circulation because the tops of the headers were straight and horizontal, which caused some of the water which would like to go upward, as it came from the tubes, to turn and move horizontally. Likewise, as it descended from the back end of the drum it was required to move horizontally.

It did not seem to Mr. Dean that the furnace was of a nature to give good mixture of gas low in air with that having an excess of air, and certainly there was no further opportunity for a good mixture before combustion ceased. Any air that was drawn into the furnace through the hopper could not be utilized because it immediately turned upward as it entered the furnace, and merely cooled the heating surfaces.

The author speculated as to the relative advantages of stoker and pulverized-coal firing, but he did not mention the great amount of unburned carbon in the ash of the former and the small amount in the latter, nor did he mention the great amount of coal put into the stoker furnace at the start, and the great amount left at the end of a run.

Alfred Vaksdal,[12] who opened the oral discussion, said that three years ago his company, the Corning Glass Works, had been evaporating about eight pounds of water per pound of coal. The following year they had increased that to 9 lb. and the next year to 10 lb. But that was not all. Last year they had started out with evaporation of 11 lb. per pound of coal, and last August had increased it to 11½ lb. They had called in engineers from the different concerns from whom they had purchased equipment and had found, after all meters, gages and scales had been checked, that this performance was correct.

He had been asked how this result had been obtained, but he knew of nothing he could attribute this performance to, other than that for the last few years the company had not changed the coal; the same men were firing the boilers to day as did three years ago, and with their complete installation of recording instruments, they knew very closely what the boilers were doing at all times. The records were posted in the boiler house, and were discussed with the men, who were complimented for good work. The firemen had ever since been showing a decided interest in doing better.

The plant comprised four Edge Moor water-tube boilers equipped with Westinghouse underfeed stokers, a coal-weighing larry, Bailey flow meters and a Cochrane V-notch open metering feedwater heater.

The performance for the year 1925 was as follows:

Total boiler horsepower	1,468
Average boiler rating, per cent	141.3
Peak load for four hours, per cent	216
Coal burned per sq. ft. per hr., average, lb.	20.7
Coal burned at high rating per sq. ft. per hour, average, lb.	31.6
Coal consumed, lb.	35,261,619
Water fed to boilers, lb.	402,995,000
Water blown down, lb.	10,000,000
Net evaporation, lb.	392,995,000
Steam consumed by auxiliaries, lb.	50,995,000
Net steam generated for production, lb.	342,000,000
Actual evaporation, steam per lb. of coal, lb.	11.14
Factor of evaporation	1.047
Equivalent evaporation from and at 212 deg., lb.	11.66
Steam per lb. of coal, net generation, lb.	9.7
Average steam pressure (gage), lb.	178
Average barometer pressure, inches of mercury	29
Average temperature of feedwater, deg. fahr.	216
Average temperature of flue gases, deg. fahr.	500

HEAT BALANCE

	B.t.u.	Per cent
Heat absorbed by boilers	11,310	78
Loss due to moisture in coal	22	1.15
Loss due to hydrogen in coal	420	2.90
Loss due to heat carried away in flue gases	1,900	13
Loss due to carbon monoxide	30	0.20
Loss due to combustible in ashes	150	1.07
Loss due to moisture in air	30	0.20
Loss due to radiation and unaccounted for	638	3.48
Heating value of coal	14,500	100.00

H. A. Cozzens Jr.,[13] said that the rubber business in which he was engaged was peculiar in that goods were made during the day and vulcanized at night—that was, in most of the smaller plants.

Starting originally with the old water wheels, consequently, his company had tried a number of different schemes with more or less success. In one plant they had installed a bleeder turbine, but had found that there was no way whatever of balancing the amount of process steam needed with the electrical load. While they had never tried it, they had watched one or two plants, and believed that the best system would be to put in a high-pressure turbine and to exhaust steam, say, at 100 lb. back pressure for process requirements. The electrical load would be governed primarily on process-steam requirements and the excess power purchased.

Alphonse A. Adler[14] pointed out that it was not necessary to have a wate -tube boiler in order to get a good efficiency, and gave particulars regarding a horizontal return tubular boiler 72 in. in diameter by 18 ft. long. This boiler was set 7 ft. in the clear above the floor line and had a very low bridgewall—a maximum of 18 in. above the top of the grates. With this setting it had been found possible to burn bituminous coal practically smokelessly except for a very brief interval during firing, the coking method being used.

The results obtained with anthracite had also been encouraging. The fireman had been asked to fire first in his own way, and the CO_2 analysis in this case had shown an average of 7½ per cent. He had then been asked to build up the fire thickness until the CO_2 was a maximum, and it had then been found that an incandescent fuel bed 1¼ in. thick gave a practically continued average CO_2 of over 14 per cent, reaching as high as 16 per cent at one instant. Mr. Adler had found that on two other occasions an incandescent fuel bed of as nearly this thickness as could be estimated without accurate means of measurement, gave the highest CO_2 with practically no CO.

John A. Stevens[15] corroborated Mr. Adler's statements, citing a 7-ft. boiler 20 ft. long and set 9 ft. in the clear. E. D. Leavitt had set boilers this way thirty years before, he said, but not until recently had the general public and the engineering public come to recognize its value.

[11] Consulting Engineer, Boston, Mass. Mem. A.S.M.E.

[12] Plant Engineer, Corning Glass Works, Corning, N. Y. Mem. A.S.M.E.

[13] Mechanical Engineer, American Hard Rubber Co., New York, N. Y. Mem. A.S.M.E.

[14] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[15] Consulting Engineer, Lowell, Mass. Mem. A.S.M.E.

Transmission of Power on Oil-Engine Locomotives—II[1]

By ALPHONSE I. LIPETZ,[2] SCHENECTADY, N. Y.

Power transmission on oil-engine locomotives depends upon the type of the oil engine for the locomotive and upon the conditions under which the locomotive is going to run.

The present paper pertains to the transmission of power only, and gives a classification of power transmissions in the chronological order of their appearance in the art.

Class A comprises full-power elastic-fluid transmission in which the total output of the oil engine is transferred into elastic-fluid energy, which is in turn utilized for the propulsion of the locomotive. The efficiency of such a transmission is the product of the two separate efficiencies of the generation and utilization of energy. Various types of electric, hydraulic, pneumatic, compressed-steam, aero-steam, and compressed-exhaust-gas transmissions are described.

Class B consists of differential elastic-fluid transmissions in which the power from the oil engine is transferred directly to the driving wheels at top speeds of the locomotive with a very high efficiency, and partly directly and partly indirectly at all intermediate speeds. The theory of such transmissions is expounded and formulas for torques, speeds, and efficiencies are derived. Examples of electric, hydraulic, and pneumatic transmissions of this type are described, and results so far obtained are given.

Class C pertains to mechanical (gear-clutch) and direct transmissions.

The drawing of conclusions as to the possible fields of application of various types of transmissions is postponed until the questions of the types of oil engines and different conditions of locomotive performance on railroads are discussed in a separate paper.

CLASS B—DIFFERENTIAL ELASTIC-FLUID TRANSMISSIONS

In the so-called full-power transmissions mentioned in Part I, the total output of the prime mover is always transmitted to the driving wheels through an elastic medium, with the result that, owing to unavoidable losses, only a part of the transferred energy produces useful work. This loss of energy takes place not only at starting, or at low speeds, when a starting device or a transmission is in any event necessary, but also at speeds and torques which otherwise could be easily obtained from the oil engine if it were directly connected with the driving wheels. A full-power transmission would seem to be an unsuitable device for speeds at which it becomes possible to employ the direct drive, as the latter would give a considerable gain in efficiency. Moreover, a main-line locomotive with infrequent stops would, most of the time, run at full speed and would require a comparatively small torque, and only occasionally at starting and on long grades would it require the help of a transmission. It would therefore seem a pity to use a transmission all the time for no other reason than because, for a very small portion of the time, the transmission is indispensable.

A. E. L. Chorlton, of William Beardmore & Co., Glasgow, Scotland, developed a design of a 2-10-2 internal-combustion locomotive consisting of a six-cylinder V-type oil engine with a mechanical transmission and a hydraulic clutch between the engine and the driving wheels. The mechanical transmission provides the increase in torque at starting and permits a direct drive when a certain speed has been attained.[3] The details of the design, however, have never been revealed. A similar scheme has been recently devised by Prof. C. A. Norman. He uses a planetary gear with a hydraulic clutch, which permits a considerable increase in torque at starting with a direct drive at certain speeds.

An excellent solution of the problem is offered by differential transmissions in which a certain portion of the power is transmitted by fluid and the remainder is carried through mechanically, the proportion of the latter part varying from 0 to 1 with the increase of velocity from nothing to full speed. Many designs of this kind have been suggested in the last 20 to 25 years, mostly for automobiles, and some of them are now being developed for locomotives.

The designs of these transmissions are sometimes complicated and difficult to conceive, but the underlying idea is simple, and is helpful to the proper understanding of the most complex designs.

Let us first consider an ordinary friction clutch inserted between the prime mover P keyed to the primary (driving) shaft A (Fig. 8), and the secondary (driven) shaft B connected to the driving wheels. When cone R is pressed against cone C, torque T_1 is transmitted to shaft B by direct contact and with practically no loss in value. With respect to power, however, great loss takes place during the whole period of acceleration. If we denote the speeds of shafts A and B by n_1 and n_2, the respective powers will be in certain units $T_1 n_1$ and $T_1 n_2$, and the loss will be equal to $T_1(n_1 - n_2)$. At the moment when shaft B acquires the speed of shaft A, the loss vanishes and the whole mechanism—prime mover, shafts, and clutch—rotates as one unit. From starting to that moment, however, a certain amount of work equal to $\sum_{n_2=0}^{n_2=n_1} T_1(n - n_2)\,\Delta t$, where Δt is the decrement of time, is wasted in destroying the friction surfaces of the clutch.

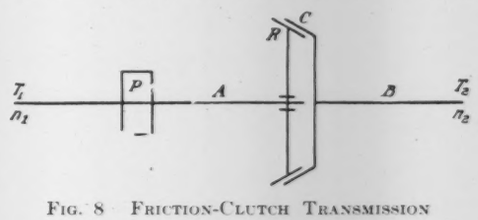

FIG. 8 FRICTION-CLUTCH TRANSMISSION

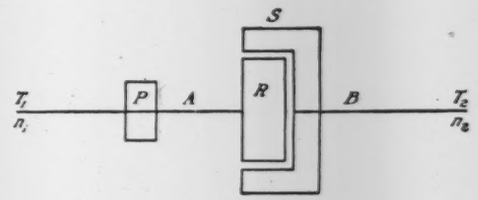

FIG. 9 HYDRAULIC-CLUTCH TRANSMISSION

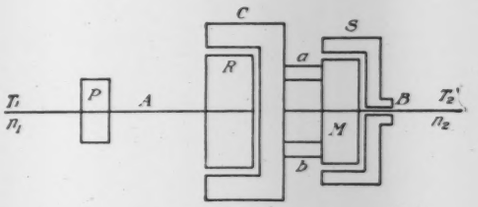

FIG. 10 DIFFERENTIAL FLUID TRANSMISSION

Imagine now a hydraulic pump instead of a friction clutch, the rotating part R (Fig. 9) of the pump being connected to the primary shaft A and the stationary part S to the secondary shaft. At starting, when S and B are at standstill, prime mover P will generate pressure in the pump until the pressure torque balances torque T_1. Then, if this pressure torque is sufficiently large to overcome the resistance torque of starting T_2, it will tend to rotate shaft B with an increasing speed until the speeds of both shafts are practically equal. During the acceleration period, however, the torque applied to S being equal to torque T_1 applied to R, speeds n_1 and n_2, however, being different, the power transmitted to shaft B is smaller than that of the prime mover applied to A. The difference $T_1(n_1 - n_2)$ is the same as in the case with the friction clutch, but contrary to the latter, the energy lost is not wasted in destroying the metal of the surfaces, but in leakage and in heating by friction. The resulting transmission is nevertheless the same, and this sort of transmission is rightly called a "hydraulic clutch."

BASIS OF DIFFERENTIAL-FLUID TRANSMISSION

Now, let us see whether it would not be possible to utilize the wasted energy $T_1(n_1 - n_2)$. Suppose we take a hydraulic clutch and, in addition to it, set a motor on shaft B (Fig. 10). Suppose

[1] Presented at the Spring Meeting, San Francisco, Cal., June 28 to July 1, 1926, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Part I, dealing with Class A transmissions, appeared in the August issue.

[2] Consulting Engineer, American Locomotive Co. Mem. A.S.M.E.

[3] The Internal Combustion Locomotive, by J. S. Tritton. Journal of the Institution of Locomotive Engineers, London, May–June, 1923, p. 367.

further that we lead the excess fluid under pressure from pump RC into the motor, coupling rigidly casing C of the pump with rotor M, while casing S of the motor is made immovable. Passages a and b are pressure and discharge pipes between the pump and the motor. Denote the arms on which the pressures in the pump and in the motor are applied by r_1 and r_2. (If, for instance, the pump and the motor are of the cylinder-plunger type, r_1 and r_2 are the crank radii of the pump and motor, respectively, multiplied by the number of cylinders.) Suppose further that prime mover P drives shaft A with torque T_1 and speed n_1, developing power N, equal in certain units to

$$N = T_1 n_1$$

and that shaft B is stationary, ready for starting. Then the pressure p that will be established in the pump must satisfy the equation

$$T_1 = pr_1 \ldots [1]$$

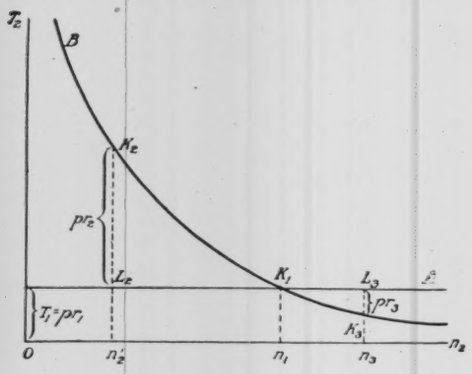

FIG. 11 TORQUE CURVES

The existing pressure between R and C will tend to rotate C and shaft B with two torques: one applied to C which is the direct action of pressure from T_1 and is equal to it, and another applied to M, equal to pr_2. The combined torque T_2 will thus be

$$T_2 = T_1 + pr_2 \ldots [2]$$

which is larger than torque T_1 transmitted in the case of the friction or hydraulic clutch. If T_2 is sufficiently large to overcome the resistance of starting, shaft B will start rotating and gradually acquire a speed n_2, which will be easily found as follows: Denote power transmitted through C by N_d; then $N_d = T_1 n_2$; denote further power transmitted through M by N_h; it equals torque pr_2 times speed n_2, whence $N_h = pr_2 n_2$. If we disregard for the moment the losses in the whole transmission, the two powers combined must be equal to N; thus

$$N = T_1 n_1 = N_d + N_h = T_1 n_2 + pr_2 n_2 \ldots [3]$$

or

$$n_2 = \frac{T_1 n_1}{T_1 + pr_2} \ldots [4]$$

This formula determines the speed n_2 that will be maintained as long as the resistance torque equals T_2 (eccentricity of the motor equals r_2—see below).

Note also that the denominator in [4] is nothing else than torque T_2; consequently

$$n_2 = \frac{T_1 n_1}{T_2}$$

or

$$T_2 n_2 = T_1 n_1 = N \ldots [5]$$

This is another expression for the fact that the power of the prime mover is transmitted completely, if the ordinary losses are temporarily disregarded.

A graphical study might better elucidate the subject treated. Assuming that T_1, n_1, and N of the prime mover are constant, we obtain a hyperbolic relation $T_2 n_2$ = constant between torque T_2 and speed n_2 of the secondary shaft represented by curve B (Fig. 11), the ordinates of which give torques on shaft B at various speeds. With an ordinary friction or hydraulic clutch we should have a straight line A giving a constant torque T_1. The excess torque which we get for any speed n_2' between 0 and n_1 is represented by $K_2 L_2 = pr_2$. It is proportional to r_2, p being equal to T_1/r_1 and consequently constant. It follows that the eccentricity of the motor must be made variable and controllable in accordance with the speed. It can be readily derived from formulas [1], [2], and [5] that

$$r_2 = r_1 \left(\frac{n_1}{n_2} - 1 \right) \ldots [6]$$

The higher the speed n_2 of the secondary shaft, the smaller must be the eccentricity. When r_2 is made equal to zero, $n_2 = n_1$, and the secondary shaft rotates with the same speed as the primary shaft. What actually happens is this: The eccentricity is zero, motor M is in no position to receive fluid under pressure (for instance, in the cylinder-piston type there is no relative movement between cylinder and piston and no room for the fluid to enter), the total quantity of fluid is locked in between R and C, and both shafts with all movable parts of the pump and motor are rotating as one solid unit. This condition is known as a "hydraulic lock." If there were cocks on pipes a and b they might just as well be closed. The torque at that speed is T_1 (point K_1) and the whole power is transmitted directly from R to C, and none is transmitted hydraulically, that is,

$$N = N_d; \; N_h = 0$$

On a chart (Fig. 12) showing the powers N_1, N_d, and N_h for various speeds, the hydraulic-lock condition is represented by points Q and U. At starting all the power is transmitted hydraulically and none directly, or

$$N = N_h; \; N_d = 0$$

This is represented by points V and O. For all intermediate speeds the straight lines OQ and VU represent respectively the power transmitted directly, N_d, and hydraulically, N_h. It is obvious that for all speeds $N_d + N_h = N$.

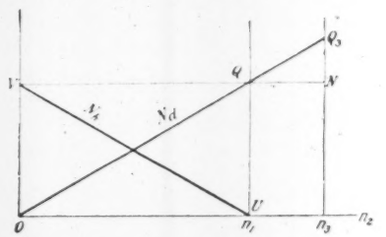

FIG. 12 POWER FOR VARIOUS SPEEDS

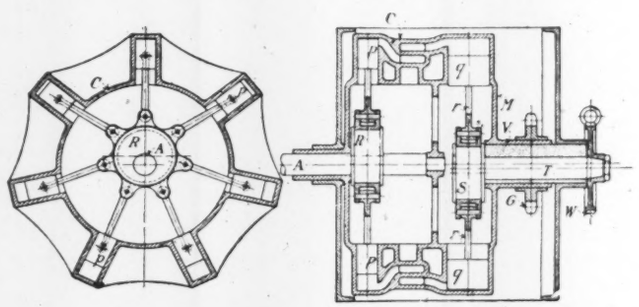

FIG. 13 THE HALL TRANSMISSION

THE HALL TRANSMISSION

Before pursuing any further the theory of the differential fluid transmission, the author believes that an example of an actual design would be very elucidating. The Hall transmission will serve the purpose best. Fig. 13 represents two vertical cross-sections through the transmission. A is the primary shaft; it drives eccentric R of a multi-cylinder piston pump, the cylinder block C of which is geared to the secondary shaft (not shown on the drawing) by means of wheel G. Cylinder block C constitutes one unit with a multi-cylinder hydraulic motor M, the pistons of which, together with their rods r and eccentric strap s, can rotate around a fixed eccentric S. The actual eccentricity of the latter in relation to the axis of the cylinder block is controlled by a wheel W keyed to a shaft T which can be turned in an eccentric bushing V. The eccentricity of the latter is equal to that of eccentric S. By turning it with shaft T in bushing V, eccentric S can be placed either in line with the prime-mover shaft and the cylinder block, in the other extreme, or in any intermediate position.

The operation of the transmission is as follows: Before starting,

shaft A rotates at full speed and drives eccentric R, which moves pistons p in cylinder block C. Motor M with pistons q and gear G is at a standstill. The primary part will now become a full-stroke piston pump generating pressure inside the cylinder block. As the fluid has no other means of coming out than through the cylinders of the secondary part, it will push pistons q as soon as the pressure is sufficient to overcome the resistance of starting. It will then turn wheel G with everything geared to it, thus establishing a continuous flow of liquid through the pump and cylinders. On the other hand, as the speed of wheel G, due to the acceleration of the locomotive, goes up, the relative speed between shaft A and cylinder C of the pump slows down. Let us assume that the cylinder block with wheel G attains a speed of one-sixth that of shaft A. The relative speed of the pump will be only five-sixths of the speed of A, and only five-sixths of the full output of the pump will be discharged into the motor. The cylinders of the motor must thus absorb five-sixths of the full output of the pump, or, as wheel G rotates six times slower than shaft A, the cylinders of motor M must, during one-sixth of a revolution, sweep a volume five-sixths of the output of the pump, or, during a whole revolution, a volume five times that of the pump. Eccentric disk S must therefore be placed in such a position as to provide the necessary stroke for pistons q. The torque transmitted through motor M is thus five times as large as that directly applied through cylinder block C, and the total torque on wheel G is six times that of shaft A—this as had to be expected, since the speeds are in a ratio of 6 to 1.

As the speed of wheel G increases, the pump delivers less and less fluid. In order to keep the pressure constant and thus balance the constant torque of the engine, it is necessary to turn disk S in accordance with the reduced pump delivery. If we reduce the secondary eccentricity too much we increase the fluid pressure and the torque, and overload the oil engine; conversely, if we increase the secondary eccentricity of S, the events will be reversed and will result in underloading the oil engine. In order to obtain the full power of the oil engine at all speeds, it is necessary to control the eccentricity of disk S in strict accordance with the variation in speed and torque. This can be arranged automatically from the pressure in the pump, and devices of this sort have already been built in connection with a gear of a certain type.

From the analysis of the differential transmission given above it is clear that a differential transmission, like the full-power transmission, must have two parts, the primary and the secondary, each of which has two members, or four members altogether. For future convenience we may call them the first, second, third, and fourth members. In a full-power transmission the first is the driving member, the third is the driven member, while the second and fourth are stationary. In a differential power transmission the second and third members, positively connected together (C and M in Figs. 10 and 13), constitute the driven member, the first being the driving and the fourth the stationary member.

Efficiency of Differential Fluid Transmission

Now we go to the question of efficiency. We assumed above that the total power is transferred without any losses from the primary to the secondary shaft. In actual practice, of course, there are unavoidable losses both of the power transmitted by hydraulic action through the pump and the motor, and of the power carried over directly by static pressure of the fluid. Thus

$$N_2 = N - N' - N'' = (N_h - N') + (N_d - N'') \quad [7]$$

where N' is the losses in the pump and motor like those in any ordinary hydraulic transmission, and N'' the losses due to static pressure, mostly leakage. If we denote the efficiency of the former by η_h and that of the latter by η_d, that is,

$$\frac{N_h - N'}{N_h} = 1 - \frac{N'}{N_h} = \eta_h$$

$$\frac{N_d - N''}{N_d} = 1 - \frac{N''}{N_d} = \eta_d$$

the power obtained on the secondary shaft will be

$$N_2 = \eta_h N_h + \eta_d N_d,$$

and the total efficiency of the transmission can be expressed by the formula

$$\eta = \frac{N_2}{N} = \frac{\eta_h N_h + \eta_d N_d}{N} \quad [8]$$

Referring to Fig. 12, we see that

$$N_h = N\left(1 - \frac{n_2}{n_1}\right); \quad N_d = N\frac{n_2}{n_1}$$

where n_2 is the variable speed of the secondary shaft and n_1 is the constant speed of the primary shaft. The efficiency is thus

$$\eta = \eta_h\left(1 - \frac{n_2}{n_1}\right) + \eta_d\frac{n_2}{n_1},$$

or

$$\eta = \eta_h + (\eta_d - \eta_h)\frac{n_2}{n_1} \quad [9]$$

Efficiency η_d is usually very high, about 90 per cent (as will be seen later in discussing the Schneider gear); η_h, as we know it from tests with full-power hydraulic transmissions, is low, at low speeds about 65 per cent and as an average, 70 per cent. Efficiency η will thus be

$$\eta = 0.7 + 0.2\frac{n_2}{n_1}$$

It is therefore always higher than the full-power transmission efficiency (70 per cent), attaining its maximum at $n_2 = n_1$, namely, 90 per cent. Formula [9], however, should not be used for speeds $n_2 > n_1$, as this case has not yet been investigated.

Conditions for High Driven-Shaft Speeds

Let us return to Fig. 11 and consider the moment when at point K_1 the actual eccentricity of S (Fig. 13) is zero and the speeds of the primary and secondary shafts are equal. If for some reason or other the speed of the secondary shaft should increase, reaching a higher value of n_3, we must at the same time move the eccentric in the opposite direction if we wish to avoid racing of the oil engine. Motor M on the secondary shaft will then work as a pump and will deliver oil back in the pump of the primary shaft, actually in the space between pistons p and cylinder block C. The pressure in this space will restrain the primary shaft in its tendency to follow the secondary shaft, and will cause it to continue to rotate at constant speed. Using the same notation as above, we may write

$$T_3 = T_1 - pr_3 \quad [2']$$

and

$$n_3 = \frac{T_1 n_1}{T_1 - pr_3} \quad [4']$$

which follow the law of the hyperbola shown in Fig. 11, as well as

$$N = T_1 n_3 - pr_3 n_3 \quad [3']$$

and

$$N = N_d - N_h \quad [3'']$$

as shown in Fig. 12, point Q_3. If we agree to consider r and N_h positive, or negative, in accordance with the position of the resultant eccentricity one way or another, and dependent upon the work of M like a motor, or a pump, formulas [2], [3], and [4] hold good for both cases.

It is different, however, with the formula for efficiency. As formerly, we shall have

$$N_3 = N - N' - N'' = (N_d - N'') + (N_h - N') \quad [7']$$

The whole power is transmitted now directly by pressure (N_d) and part of it is absorbed by working the motor as a pump (N_h being negative). We have, therefore, using the same notation for efficiencies as above,

$$\frac{N_d - N''}{N_d} = 1 - \frac{N''}{N_d} = \eta_d$$

and, N_h being negative,

$$\frac{-N_h - N'}{-N_h} = 1 + \frac{N'}{N_h} = \eta_h \ldots\ldots\ldots [10]$$

formula [7'] can be transformed into

$$\eta = \frac{N_3}{N} = \frac{N_d - N''}{N_d} \cdot \frac{N_d}{N} + \frac{N_h - N'}{N_h} \cdot \frac{N_h}{N}$$

But

$$\frac{N_d}{N} = \frac{n_3}{n_1}; \qquad \frac{N_h}{N} = 1 - \frac{n_3}{n_1}$$

and in accordance with [10],

$$\frac{N_h - N'}{N_h} = 1 - \frac{N'}{N_h} = 2 - \eta_h$$

therefore

$$\eta = \eta_d \frac{n_3}{n_1} + (2 - \eta_h) \cdot \left(1 - \frac{n_3}{n_1}\right)$$

or

$$\eta = 2 - \eta_h - \frac{n_3}{n_1}(2 - \eta_h - \eta_d) \ldots\ldots\ldots [11]$$

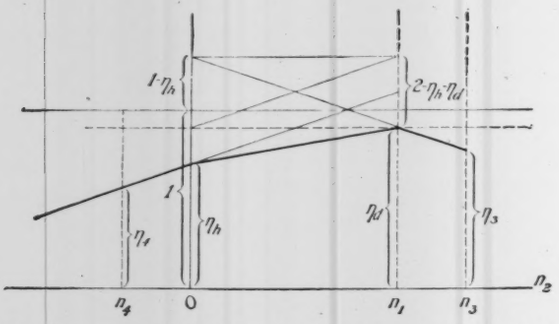

FIG. 14 EFFICIENCY CURVE, DIFFERENTIAL HYDRAULIC TRANSMISSION

If we assume $n_3 = n_1$, this formula renders

$$\eta = \eta_d$$

or the same as formula [9] for $n_2 = n_1$. This is quite natural.

We have thus two formulas for efficiency η:

for $0 < n_2 < n_1$, $\quad \eta = \eta_h + (\eta_d - \eta_h)\frac{n_2}{n_1} \ldots\ldots\ldots [9]$

for $n_3 > n_1$, $\quad \eta = 2 - \eta_h - (2 - \eta_h - \eta_d)\frac{n_3}{n_1} \ldots\ldots\ldots [11]$

Fig. 14 represents them graphically and Fig. 15 shows efficiency curves obtained from actual tests with the Schneider gear, which is a typical differential hydraulic transmission (see below). The difference in slope of the two branches of the efficiency curve, for speeds below and above the oil-engine speed, can be easily seen.

The efficiency drops very rapidly on the second branch of the curve. Theoretically it will become zero at

$$n_3 = \frac{2 - \eta_h}{2 - \eta_h - \eta_d} n_1$$

If we use the numerical values above given for η_h and η_d (0.70 and 0.90), this critical speed becomes equal to 3.25 times n_1. Differential elastic-fluid transmissions are never designed for speeds above $1.5n_1$, at which speed the transmission can still render an efficiency of 65 to 70 per cent. Higher speeds should not be attempted with transmissions of this type.

The efficiency curves representing our formulas are straight lines, as we assumed that η_h and η_d are independent of speed. Actually they depend upon the relative piston speed, and therefore their real form is as shown in Fig. 15. The critical-speed value ($3.25n_1$), at which $\eta = 0$, has no practical value and was used only as an illustration.

REVERSAL

Let us again return to Fig. 13. With r_3 negative and approaching r_1, we obtain ever-increasing speeds and, theoretically speaking, when r_3 becomes equal to r_1, the speed becomes infinity. Let us now investigate what will happen if r_3, being still kept negative, is made larger than r_1. Imagine the starting of the transmission with the eccentric beyond its extreme negative position. Gear G, motor M, with cylinder block C, are at standstill. Rotor R will then act as a pump and deliver fluid under pressure to motor M. Similar to what took place when the eccentric was in its extreme positive position, the motor, at the moment when the pressure becomes sufficiently high to overcome the resistance of starting, will begin to turn, but in the opposite direction. The rotation of C will then be opposite to R and the output of the pump will increase until a state of stabilization is established. This will take place when motor M is in a position to absorb the amount of oil delivered by the pump. The speed of the secondary shaft n_4 will be such that (r_4 being the negative eccentricity)

$$(n_1 + n_4)r_1 = n_4 r_4$$

or

$$-r_4 = r_1\left(\frac{n_1}{-n_4} - 1\right)$$

which is identical with formula [6], the difference being only in the signs of r_4 and n_4 which are negative.

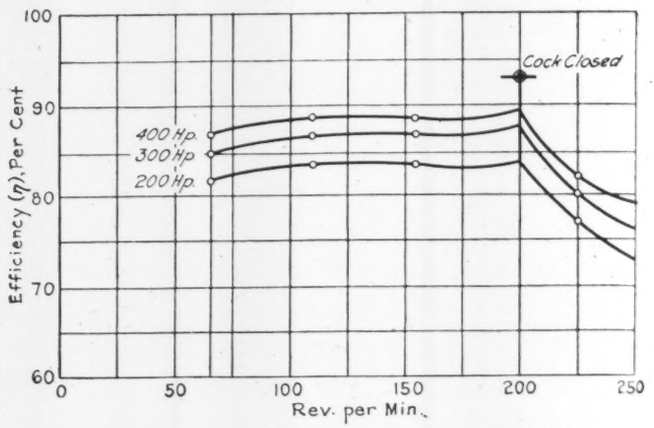

FIG. 15 ACTUAL EFFICIENCIES, DIFFERENTIAL HYDRAULIC TRANSMISSION

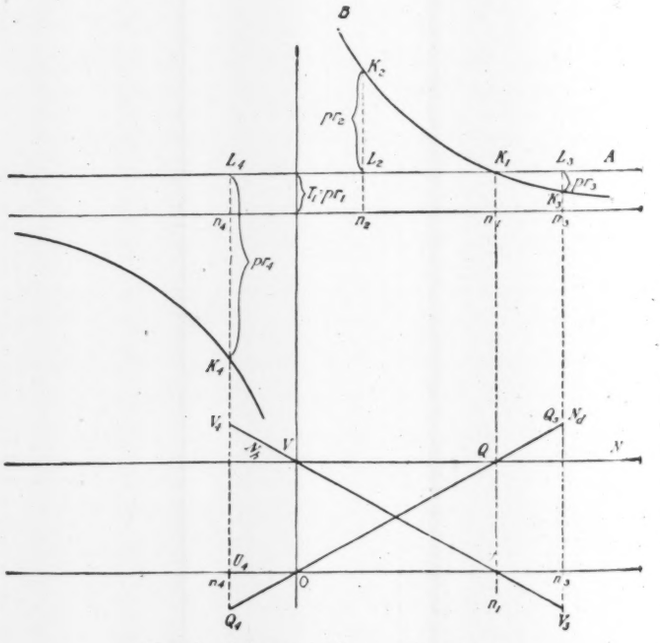

FIG. 16 REVERSED-MOTOR CONDITIONS

The torque on the secondary shaft will be a resultant of two: first, T_1, the torque of the oil engine acting in the direction of its rotation, and second, pr_4 from the hydraulic motor M acting in the opposite direction. But since r_4 is negative, we ought to write

$$T_4 = T_1 + pr_4$$

which is identical with formula [2]. We thus obtain the other branch of the hyperbola, as can be seen in Fig. 16. All other formulas also hold good. For power, for instance, we have (Fig. 16)

$$OV = V_4U_4 - U_4Q_4$$

or

$$N = N_h - (-N_d) = N_h + N_d$$

as before.

But as regards efficiency, this must be again investigated separately because losses of power are always negative. Reasoning in the same way as we did in the case $n_3 > n_1$, we obtain

$$\eta_4 = \eta_h - (2 - \eta_h - \eta_d)\frac{-n_4}{n_1} \quad \ldots\ldots\ldots\ldots [12]$$

$-n_4$ being positive. At $n_4 = 0$ we obtain $\eta_4 = \eta_h$, as it should be. In Fig. 14 the efficiency for negative speeds is shown as a straight line. It falls rapidly with the increase in the value of the speed. The inclination of the line is the same as that of line GH, which is the efficiency line for speeds over n_1. This can be inferred from the respective formulas of the two lines. Theoretically speaking, the efficiency becomes zero at speed

$$n_4 = \frac{\eta_h}{2 - \eta_h - \eta_d} \cdot n_1$$

If the former values should be applied ($\eta_h = 0.7$; $\eta_d = 0.9$), the efficiency is zero at $n_4 = 1.75n_1$. We thus see that the reverse-speed efficiency is even worse than the efficiency at high speeds ahead. It is therefore not advisable to use a transmission of this type for speeds much below zero and above n_1.

The differential elastic-fluid transmission can be materialized with any kind of fluid, and therefore there can be as many types of the differential transmission as there are of the full-power transmission. So far there have been developed electric, hydraulic, and pneumatic differential transmissions:

1—Differential Electric Transmissions

(*a*) *Entz Transmission.* About twenty years ago the Entz electric drive was brought out as a means for controlling the speed of self-propelled vehicles, more particularly automobiles. It has all the characteristics of a differential transmission. A section through an Entz automobile transmission is shown in Fig. 17. Two dynamos serve as primary and secondary units, or, in ordinary cases, as generator (on the left) and motor (on the right). The field frame of the generator (first member) is bolted to the driving shaft, the armatures of the generator (second member) and motor (third member) are, in accordance with our scheme of differential gears, fastened to the driven shaft, and the field frame of the motor (fourth member) is stationary. At starting, when the driven shaft is also stationary, the primary dynamo (on the left) acts as a full-speed generator, and the current thus generated drives the secondary dynamo (the motor) through a suitable electric control. The transmission acts at starting as a full-power electric transmission. But as the speed of the driven shaft increases, less current is generated and less energy is transmitted electrically, the remainder of the power being transferred from the driving shaft through the field coil directly to the driven shaft, so to speak, by a magnetic drag. At full speed, when the driven shaft has attained the speed of the driving shaft, the motor receives no more current, and the whole power is transmitted wholly magnetically, the generator acting as a very flexible clutch.

Simultaneously with the change in speed the motor field excitation must be gradually decreased. At full speed the generator's brushes are short-circuited. The motor, not being used at high speed, may be converted into a generator to charge the storage battery. The motor may be employed also as an electric brake. The starting of the prime mover (oil or gasoline engine) is accomplished by operating the generator as a motor from the storage battery.[4]

The efficiency of the transmission is the combined efficiency of the electric transmission and of the clutch generator. Formulas [9] and [11] hold good if η_h is the total efficiency of the electric transmission (generator and motor) and η_d that of the magnetic dragging effect.

The Entz transmission was applied to some extent in automobiles, and while its performance was fairly good, it met with very little success. The author does not know the reasons for this, but, as far as he can see, there was very little need for such a transmission for automobiles. The efficiency, while slightly higher than that of a full-power transmission, was still less than that of the ordinary clutch and gear box. In addition, the magnetic drag is in no way so positive as a hydraulic lock, and considerable slippage must have taken place at high speeds. This probably explains the fact that the oil engine had to be run at a higher speed than it would have been in a geared car moving at the same speed on the road.

The transmission was never tried for rail cars or locomotives, although it would seem that at least for rail cars, and possibly for buses, the Entz transmission would be worth trying.

(*b*) *Thomas Transmission.* As we have seen before, a differential transmission consists of four members, of which the first (driving) member runs at the prime mover's speed, the second and third members, rigidly connected together, are driven at a variable speed, and the fourth member is stationary. Thus it is a mechanism with two degrees of freedom. When the transmission is in operation the extra degree of freedom is gradually absorbed by the resistance of the fluid, until finally at top speed it is absorbed entirely.

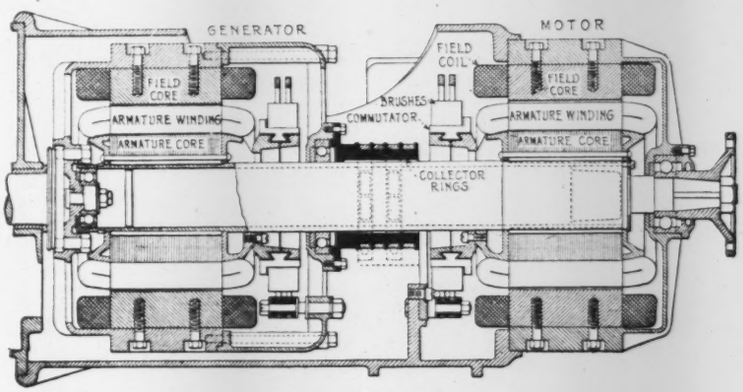

FIG. 17 THE ENTZ TRANSMISSION

The required degree of freedom can also be obtained in another way, namely, we can somewhere between the A and the B ends of the transmission interpose a planetary or a differential gear which, as is well known, has two degrees of freedom. Further, we can absorb the extra degree of freedom by an electric or hydraulic slippage at the A end which can be utilized as a booster on the B end. The two machines, however, will have both stators immovable. A good example of an electric drive based on this principle is the Thomas transmission.

A diagram of the transmission is given in Fig. 18. The engine drives the bevel wheel on the left; the arm with the two bevel pinions is connected to the driven shaft marked "load;" the right bevel wheel is attached to a hollow shaft marked "auxiliary." The speeds of the three shafts are interrelated as shown in Fig. 19. Imagine further a generator G on the auxiliary shaft and a motor M on the driven shaft (Fig. 18). At starting the resistance will keep the driven shaft at standstill, and if the engine is running forward, the generator will rotate in the opposite direction with the same speed. If we utilize the electric current for driving the motor on the load shaft, we can by using proper electric control exert

[4] *Electrical World*, vol. 63, no. 5, January 31, 1914, p. 281, and vol. 67, no. 2, January 8, 1916, p. 110. See also The Columbia Electrical Transmission, by V. W. Page. *New England Automobile Journal*, October 26, 1907.

the necessary tractive effort and overcome the resistance. As the forward speed of the load shaft increases, the backward speed of the auxiliary shaft decreases until it vanishes altogether (point *A* on Fig. 19). The load shaft is then revolving with half speed of the engine. No current is generated in *G*, and the total power of the oil engine is transmitted wholly mechanically. The torque is double that of the oil engine, disregarding, of course, the friction losses in the gear. Mr. Thomas, the inventor, calls the transmitting of power at this speed the "intermec drive." Up to this speed the power is transmitted both mechanically and electrically.

If the speed of the load shaft goes up beyond the intermec speed, the auxiliary shaft begins to rotate in the forward direction and by a simple alteration of the electrical connection, *M* is converted into a generator which delivers current into *G*, which becomes a motor. The speed of the auxiliary shaft increases, as also does

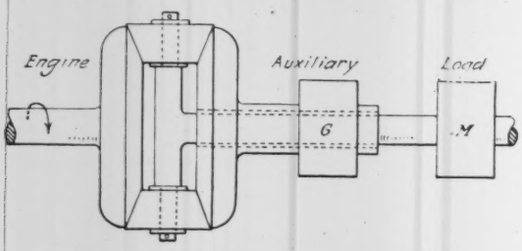

FIG. 18 PRINCIPLE OF THOMAS TRANSMISSION

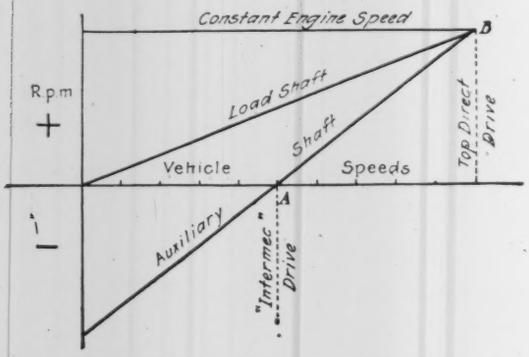

FIG. 19 SPEED DIAGRAM FOR FIG. 18

that of the load shaft—although its increase is slower than that of the auxiliary shaft—until the two speeds reach the speed of the oil engine (point *B*). Thomas calls the transmitting of power at this speed the "top direct drive." Then a mechanical coupling of any two of the three shafts by means of a clutch locks the gear, which continues to rotate as a unit with a direct transmission of power from the oil engine to the wheels. The electric drive is no longer necessary and can be cut out entirely. The losses being only those of bearing friction and air resistance, the efficiency is very high.

The bevel-gear form of the Thomas differential transmission is not an essential feature of the transmission. This form has been used as an illustration in order to follow the speed relations. Actually, the Thomas transmission is built with a planetary gear as shown in Fig. 20. Casing *C* takes the place of the arm and auxiliary shaft of the differential gear; wheels *P* are the planetary wheels, the two sun wheels being fastened to the driving and to the driven shafts. The advantages of such an arrangement are that the bevel wheels are omitted and that the gear ratio does not necessarily have to be 1 to 2. The intermec speed may therefore be made less or more than one-half of the top speed.

One difference between the Thomas and the Entz transmission is that in the former the magnetic drag of the latter is eliminated and a positive transmission through toothed wheels takes the place of the uncertain magnetic drag. Another difference is that the Thomas transmission has two speeds at which the power is transmitted purely mechanically, those of the intermec drive and of the top direct drive, whereas the Entz gear transmits the power this way only at the top speed, and then does it, so to speak, pseudomechanically, by magnetic drag.

These two essential differences are reflected in the efficiency curves. Fig. 21 gives the calculated efficiency of the Thomas transmission. If compared with Fig. 15, showing the actual efficiency of the Schneider gear, the difference in the characters of the efficiency curves is evident. No attempt, however, should be made to compare the efficiency figures of the two gears, as the Thomas transmission figures are calculated while the others are actual.

The Thomas transmission was applied for the first time some thirteen or fourteen years ago to a 36-hp. Leyland truck in England. The truck was tested and the performance found to be satisfactory. Later a small Delahaye car weighing 3800 lb. was equipped with

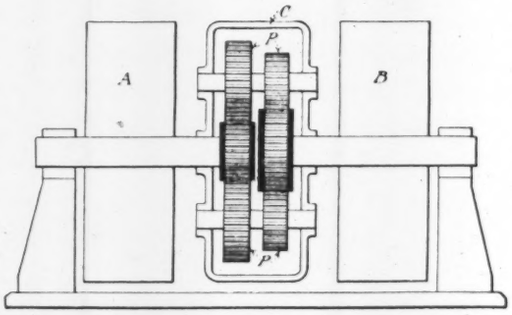

FIG. 20 DIAGRAM OF THOMAS TRANSMISSION SYSTEM

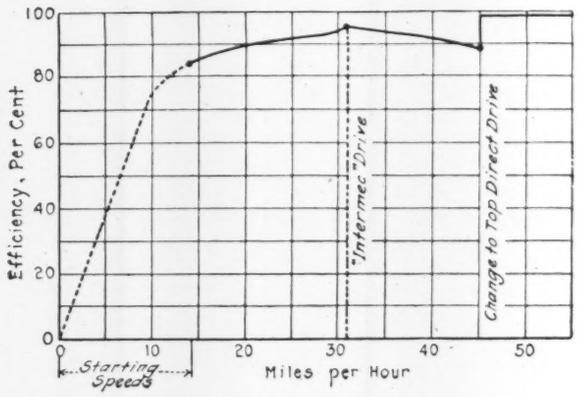

FIG. 21 EFFICIENCY CURVE OF THOMAS TRANSMISSION

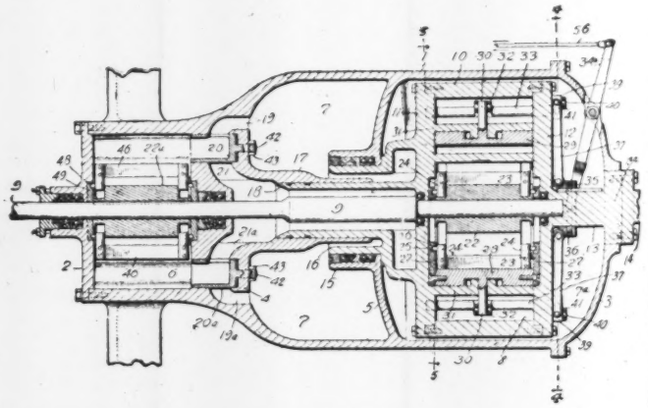

FIG. 22 RAYBURN TRANSMISSION

a Thomas transmission. Mr. Thomas reports that on a run from London to Edinburgh and back the petrol consumption was an imperial gallon per 36 miles of run.[5] Several other cars and buses were built with the Thomas transmission, and in 1914 a 160-hp. rail car weighing 26.3 short tons was built for South Africa. An official test made both ways between Pretoria and Pienarr's River over a distance of 84.6 miles (total run 169.2 miles) showed an average fuel consumption of an imperial gallon per 8.65 miles or

[5] The Thomas Transmission, by Hedley J. Thomson, p. 17; paper read before the New Zealand Society of Civil Engineers at Dunedin on March 11, 1915.

228 ton-miles.[6] This corresponds to 3.5 lb. of gasoline per 100 ton-miles. In 1915 another rail car with an eight-cylinder 200-hp. V-type engine and Thomas transmission was built for New Zealand, but no information as to the performance of this car is available. Nor is there any published information to be found as to the further progress made by the Thomas transmission, although designs of 1000 to 1200-hp. locomotives have been worked out.[7]

2—Differential Hydraulic Transmissions

Similarly to the full-power transmission, the differential hydraulic transmission also permits of a great variety in design on account of the great number of different types of hydraulic pumps and motors in existence. In addition, two different classes are possible, without or with the planetary gear, corresponding to either the Thomas or the Entz transmission. This explains possibly the fact that for the last three decades the domain of differential hydraulic transmissions, if judged by the number of patents issued, has been one of the most prolific in mechanical engineering. The majority of the transmissions were intended for automobiles, but several of them were also proposed for locomotive drive.

(*a*) *Rayburn Gear.* The Rayburn gear is the hydraulic analogy of the Entz transmission. Oil-engine shaft 14 (Fig. 22)[8] is directly attached to the first member consisting of casing 10 with heads 11 and 12. The second member consists of a rotor 22 with blades 23. The two members constitute together a pump of the vane type. The third member consists of a similar rotor 22*a* with blades 46. Rotor 22*a* is keyed to shaft 9, to which rotor 22 is also fastened. Shaft 9 is journaled in a stationary head 2 and in heads 11 and 12. The fourth member is stationary and forms a part of casing 1. The third and fourth members represent a motor also of the vane type which takes the place of the booster unit of the Entz transmission. Vanes 23 and 46 can be moved toward and away from the axis of shaft 9; this movement is controlled from outside.

As regards the operation of the transmission, one can readily see that at starting the two rotors are at standstill. The oil engine drives the outside casing of the pump at full speed, and with the blades accordingly moved, pumping action begins in the primary part (pump on the right). Fluid under pressure passes through suitable channels and valves into the secondary part (booster unit) and tends to rotate rotor 22*a*, and with it shaft 9 and rotor 22, in the direction of rotation of casing 10. As the speed of rotor 22 increases, the pumping action of the pump decreases, and at a certain speed it disappears altogether. This latter event corresponds to the top speed, when the hydraulic lock takes place. At that speed the total power of the oil engine is transmitted to the driven shaft 9 through the hydraulic pressure in the primary part.

In order to keep the pressures in conformity with the torque of the engine at all speeds, the projecting of the blades 23 and 46 is accordingly controlled by a handle.

(*b*) *Hydraulic Analogy of Thomas Transmission.* A hydraulic transmission analogous to the Thomas electric transmission has been lately developed by an English firm. A planetary gear is directly connected to a gear pump, while a motor of the Williams-Janney type with a swash plate is connected to the driven shaft. The gear has not yet passed the experimental stage and, the author understands, no information is available at present.

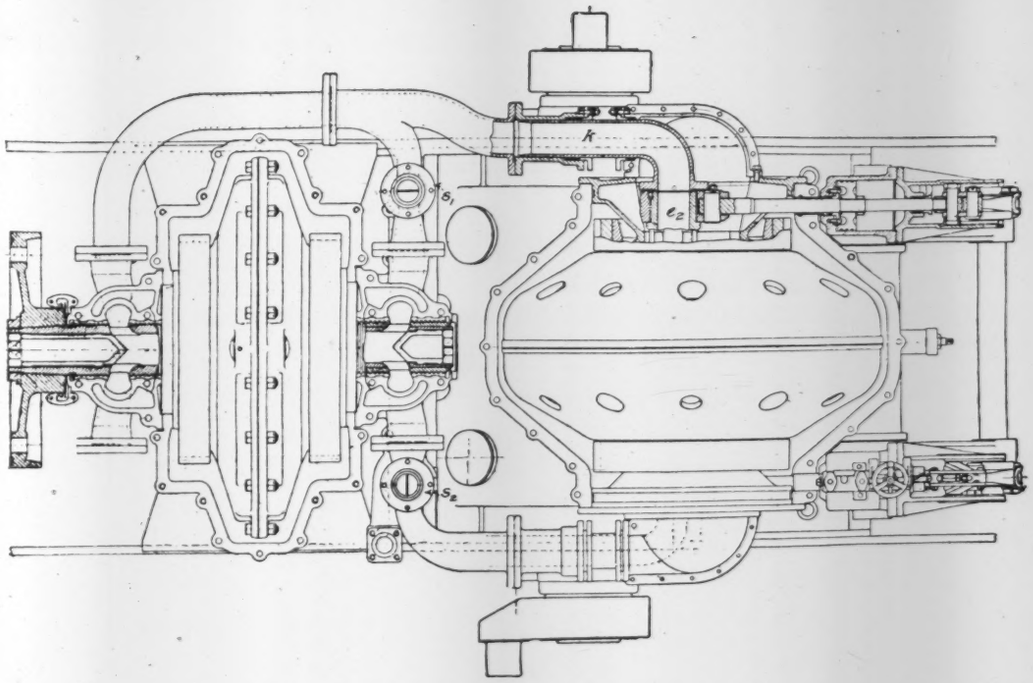

Fig. 24 Plan View of Schneider Transmission

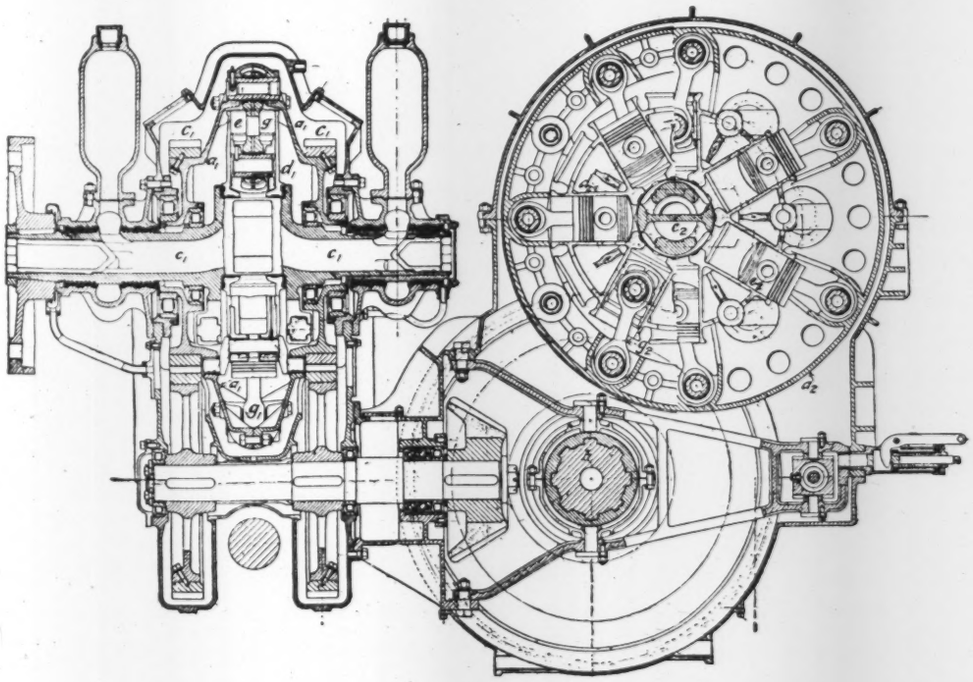

Fig. 23 Section of Schneider Transmission

[6] Ibid., p. 22.
[7] Ibid, pp. 11 and 25.
[8] U. S. Patent Specification No. 1,297,733, March 18, 1919, Fig. 3.

(*c*) *Schneider Transmission.* A very interesting modification of the Hall principle is represented in the Schneider transmission. This also consists of a primary and secondary part, the difference being that the first and fourth members are cylinder blocks instead of pistons, whereas the second and third members are rotors with pistons fitting into the cylinders. Further, the second and third members are not made as one unit, but are rigidly connected by means of gears with a driven shaft interposed between them. Fig. 23 shows a vertical section, and Fig. 24 a plan view of a 500-hp. Schneider gear built by the Swiss Locomotive Works in Winterthur, Switzerland. The oil engine drives a hollow partitioned shaft c_1 with a permanent crank operating a six-cylinder block d_1. This constitutes the first member, the constant-speed member of the primary part. The variable-speed member (second member) of the primary part consists of a rotor a_1, rods g_1 and pistons e_1. The rotor has two spur-wheel rims C_1 which mesh with two spur wheels set on an intermediate shaft connected with the driven shaft h by means of bevel wheels. Shaft h is in a similar way geared to rotor a_2, which with corresponding rods and pistons constitutes the variable-speed member of the secondary part (third member). The design of the latter is similar to that of the second member, the only difference being the double number of pistons, namely, two rows of six pistons each. The fourth (stationary) member consists of a corresponding twelve-cylinder block d_2 which can rotate around a hollow stationary shaft c_2 with variable eccentricity in relation to rotor a_2. The two shafts also perform the function of valves, establishing the proper connection between suction and delivery passages. The variation of eccentricity of shaft c_2 is effected by a power reverse which can be seen in Fig. 24. Two pistons of the power reverse act directly on shaft c_2 which is guided by pipes k sliding inside stationary suction and delivery pipes.

From what has already been said, it is easy to follow the process which takes place in a Schneider gear. Before starting, when the oil engine is idling, valve s_2 is closed and s_1 is open, and shaft c_2 is placed in its extreme position, driven shaft h being at a standstill, both rotors a_1 and a_2 with their appertaining bevel and spur wheels are stationary. The first member rotates at the speed of the prime mover, but no pressure is generated, as the oil circulates freely through valve s_1 from the delivery into the discharge passages of the primary part. If we gradually close valve s_1 and open valve s_2 oil is then forced into the secondary part where it acts against the resistance of starting; the pressure thus increases. If the eccentricity of the stationary shaft c_2 was properly chosen, the maximum torque will be sufficient to overcome the resistance torque, and rotor a_2 with shaft h and rotor a_1 will start turning, the latter in the direction of the prime mover. The relative speed between the first and second member drops and the amount of oil pumped by the primary part decreases, but the pressure remains constant if the eccentricity of the secondary part is correspondingly reduced. The resulting torque varies inversely to the speed of shaft h until it balances the resisting torque. The running of the locomotive will then continue at a speed corresponding to these balanced conditions.

When the resisting torque drops to a value in the neighborhood of the oil-engine torque, shaft c_2 is placed concentrically with rotor a_2 of the secondary part. This establishes the so-called hydraulic-lock condition, at which the total output of the oil engine is transmitted directly to shaft h through the oil in the primary part and the gear. The secondary part delivers no power, and valve s_2 can be closed.

Further increase in speed of the locomotive can be obtained either by speeding up the engine itself, or by placing shaft c_2 in a negative eccentric position, as was explained in discussing the Hall transmission.

A Schneider transmission was built by the Swiss Locomotive Works for a 500-hp. Diesel locomotive, and on May 13 and 14, 1924, it was officially tested by Prof. P. Ostertag of the Winterthur Polytechnic Institute and a full report published in March, 1925.[9] The oil engine ran at a speed of 350 r.p.m. and the driven shaft was operated at 5 different speeds: 64, 100, 150, 200, and 220 r.p.m., the 200-r.p.m. speed corresponding to the direct drive. The transmission was tested up to 400 hp. on shaft h, further increase not being possible on account of the limitation of the two Prony brakes applied during the test. Fig. 15 gives the efficiency curves for speeds between 64 and 250 r.p.m. of the driven shaft. For 400 hp. and speeds below that of direct drive, or top speed, to use Mr. Thomas' designation, the efficiency fluctuates between 87 and 89 per cent. For smaller powers (200 and 300 b.hp.) it drops to 82 per cent, and for powers below 140 b.hp. even 72 per cent was recorded. The efficiency dropped abruptly with an increase of speed beyond the top speed, this in accordance with the formulas and charts previously given. When valve s_2 was closed at top speed the efficiency went up to 93 per cent.

As regards the performance of the transmission as a mechanism, it was quite satisfactory from the start, requiring very few changes or improvements. Some leakage of oil through the working parts became noticeable and a small pump was applied in order to return the oil to the suction passage of the primary part. The power absorbed by the pump did not exceed 6.5 hp., and this amount was deducted from the net outputs when the efficiencies were computed.

The 500-hp. locomotive, however, has not yet been built, but for reasons which have nothing to do with the Schneider transmission, although some recent publications have reported its completion.

(*d*) *Gyroscopic Transmission.* Before closing this chapter on differential hydraulic gears, the author wishes to mention a very ingenious device, the so-called gyroscopic transmission of Emil Faure of Paris. This also consists of a primary and a secondary part, both of a singular design called by the inventor "gyroscopic pump" and "gyroscopic motor," respectively. A 30-hp. pump unit was built and tested, but so far very little progress has been made. The description of the pump can be found in U. S. Patent No. 1,402,853, dated January 5, 1921.

3—Differential Pneumatic Transmission

Hautier Transmission. About 15 years ago Camille J. Hautier, of Paris, patented a pneumatic transmission with a differential gear which is analogous to the Thomas transmission. A small 20-ton locomotive equipped with a 70-hp. engine was built by the firm of Schneider & Co., of Creusot, France, and has been in service since 1912.[10] It was tested in 1913 on the French State Railways with a dynamometer car. At 14.2 miles per hour, corresponding to the top speed of the transmission, the efficiency of the latter was 90 per cent; at 9.3 miles per hour it dropped to 80 per cent; at 4.3 miles per hour it was only 67 per cent. The low efficiency figures at lower speeds were undoubtedly due to the losses in the air transmission.

CLASS C—MECHANICAL TRANSMISSIONS

The great success of Stephenson's *Rocket*, the prototype of the present-day steam locomotive, must to a very large extent be attributed to the fortunate idea of a direct connection between the steam pistons and wheels. We do not know whether George Stephenson was conscious of the fact that steam, on account of its wonderful flexible properties, does not require, even for variable loads, any other transmission than the direct connection. We know, though, that others were not aware of this fact and that no less a genius than James Watt thought that a variable transmission was indispensable for a steam locomotive. In the description of his invention of a steam locomotive patented in 1784, Watt described a sun-and-planet-wheel transmission and two sets of gears, differing in their proportions, which he provided in order to adapt the power of the locomotive to the variable resistance of the road.[11]

However, Watt did not attempt to build a locomotive, either with the double or with any other transmission, as he very soon lost confidence in the possibility of propelling locomotives with noncondensing engines; and it was left to George Stephenson to prove that it could be done. Stephenson discarded a good many of the

[9] *Schweizerische Bauzeitung*, vol. 85, nos. 10 and 12, pp. 123–127 and 154–155. See also *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 16, April 18, 1925, pp. 499–504 and no. 18, May 2, 1925, pp. 595–600.

[10] Application du Moteur à Hydrocarbures à la Traction sur Voies ferrées, by M. E. Brillié, *Revue Générale des Chemins de Fer*, May, 1923, pp. 403–406. See also The Internal-Combustion Locomotive, by J. W. Hobson. Paper read before the North East Coast Institution of Engineers and Shipbuilders, February, 1925, p. 37.

[11] The World's Railway, by J. G. Pangborn, p. 14.

prejudices of his age, among them the fear of direct transmission. He tried it, succeeded, and established it as a typical locomotive feature. It is therefore quite natural that present-day locomotive designers are striving to preserve the simplicity of the direct-driven steam locomotive for the oil-engine locomotive. In order to do this they have had to solve the problem of starting and, to a certain extent, that of speed variation. These have not yet been satisfactorily solved, but very interesting attempts to do so have already been made.

The problem of starting can also be solved, especially for small power, in the same way as it is done in automobiles, namely, by starting the oil-engine without load and putting the load on by means of a clutch. This procedure has been considered impracticable for large power, but quite recently the principle has been embodied in a locomotive which will be described later.

We thus see that two possibilities present themselves: one a mechanical transmission with an interposed clutch similar in principle to the ordinary automobile drive, and the other a rigid connection between engine and wheels by means of gears, or rods, or both.

1—Gear-Clutch Transmission

Gear-clutch transmissions have been in use for a long time in small industrial locomotives and rail cars, mostly in France and in Germany for powers ranging from 30 to 200 hp. They represent a combination of clutches, of a set of gears of the sliding type, of universal joints, and bevel gears. Sometimes chains, jackshafts, and rods are also used. The well-known Renault rail cars and the Deutz and Schneider industrial locomotives employ transmissions of this kind. In this country the McKeen car was at one time very much in vogue. The Baldwin 100-hp. gasoline locomotive, of which several hundred were built during the World War, was provided with a gear-clutch transmission.

There are, however, several types which differ slightly from the ordinary automobile transmission. For instance, the 60-hp. rail car built by Messrs. Schneider & Co. for the Paris Extra-Suburban and French State Railways, is equipped with a special slipping clutch designed by Fieux, in addition to the ordinary friction clutch. This clutch is interposed between the engine shaft and the friction clutch, and serves as an automatic "connector" at and above a certain speed, and as an automatic "disconnector" below another lower speed. It is based on the action of a spiral spring pressed against the inside surface of a drum by the centrifugal force of the rotating masses connected to the oil-engine shaft. The frictional force between the spiral spring and the drum represents the driving effort applied to the transmission shaft.[12] The oil engine, therefore, can never get stalled irrespective of the car resistance, as at a certain speed before stalling it will be disconnected from the transmission shaft and will race up again. This assures a positive and smooth starting, provided the engine is of sufficient power to overcome the resistance.

The Fieux slipping clutch has been applied to ten Schneider 60-hp. rail cars, and also for other purposes in sizes ranging up to 180 hp.

The most notable difference in the latest designs of gear-clutch transmissions for cars and locomotives is the replacement of the shifting gear by a series of gears with a corresponding number of clutches, all gears being always in mesh. The friction clutches engage one gear at a time, all other gears remaining disengaged. The change from one speed to another is effected by slipping between parts of the engaged clutch. The clutches can be operated manually, pneumatically, hydraulically, or magnetically.

a The Allgemeine Elektricitäts Gesellschaft (A.E.G.) of Berlin is building rail cars with one or two 75-b.hp. MAN Diesel engines and with four sets of gears always in mesh and with four friction clutches—one clutch for each speed operated pneumatically.[13]

b Similarly, another firm in Berlin, the Waggonfabrik Wismar, has built a rail car with a four-gear, four-clutch transmission. The car is driven by the famous Maybach oil engine in a size developing 150 b.hp. at 1300 r.p.m. The engine is mounted on one of the two four-wheel trucks, and by means of bevel reversing gears drives a jackshaft journaled in the truck frame and connected to the driving wheels by means of rods.[14]

c In 1923 the Swiss Locomotive Works in Winterthur built a 0–4–0 switching locomotive with four pairs of gears always in mesh and four hydraulically operated friction clutches of the Meyer type (Fig. 25). One gear wheel of each pair is made of two parts which are screwed together and can rotate on the hubs of two inside disks which form the clutch. The disks and wheels have concentric tapered grooves and are pressed against the wheels by hydraulic pressure, which thus engages and disengages the clutches. The disks can slide on the keys by which they are set on the hollow shaft. Normally the disks are kept disengaged by oil which fills the lower passage of the shaft at a constant pressure; when necessary, they are engaged by admitting oil into the upper passage at a pressure in excess of that in the lower passage. The four speeds are: 5, 9½, 15 and 22 m.p.h. Bevel gears are used for reversing.[15]

The locomotive weighs 17.9 tons and is driven by a three-cylinder, four-cycle airless-injection oil engine developing 80 b.hp. at 450 r.p.m. Since its completion it has been used for switching service in the yards of the Swiss Locomotive Works, and another locomotive of a larger size with a Meyer clutch is now under consideration.

d The most daring application of a gear transmission has been recently made in a 1000-hp. Diesel locomotive which has been built in the Hohenzollern Locomotive Works in Düsseldorf, Germany.[16] The locomotive has been ordered for main-line service on the Russian Railways in competition with the 1000-b.hp. oil-electric locomotive previously described. It is of the 4–10–2 type and has a reversible 1000-b.hp. Diesel engine which acts on a jackshaft by means of three gears always in mesh with three friction clutches operated magnetically. The locomotive and its transmission are shown in Figs. 26 and 27. The oil engine drives shaft C through bevel gears M. An intermediate shaft B is permanently engaged with jackshaft A by means of gears N. Shafts C and B can be connected by clutch I and gears Z_1 (first speed), or clutch II and gears Z_2 (second speed), or clutch III and gears Z_3 (third speed). Each clutch is integral with one wheel of the corresponding pair of wheels, the other wheel being fastened to one or another intermediate shaft. A powerful friction clutch, also magnetically operated, is interposed between the oil engine and the bevel gear M. Thus only three speeds can be obtained at the normal speed of the Diesel engine—three forward and three backward—the Diesel engine being reversible. All intermediate speeds can be obtained by changing the number of revolutions of the Diesel engine itself. In changing from one speed to another, practically half of the total output of the oil engine during the whole period of acceleration will be spent on the destruction of the metal surfaces between the working parts of the respective clutches. However, the builders claim that the wear of the clutches is not appreciable due to the shortness of the acceleration period. The wear of the friction-clutch lining after 1000 engagings and 2500 miles of heavy freight service was found to be 0.011 in. Moreover the lining, which is

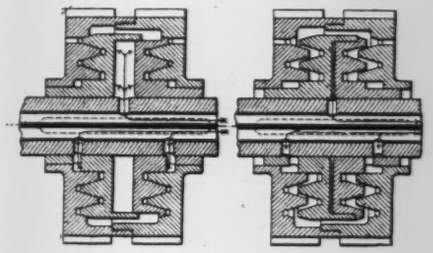

Fig. 25 Meyer Type Hydraulically Operated Friction Clutch

[12] Le Conjoncteur-Disjoncteur a Friction (Systeme Fieux), Bulletins de la Societe d'Encouragement pour l'Industrie Nationale, vol. 136, March, 1924, pp. 279–287. See also *Engineering*, vol. cxix, June 19, 1925, pp. 760–761, or *Oil Engine Power*, vol. 3, no. 8, August, 1925, p. 462.

[13] Traction on Rails by Internal-Combustion Engines, by M. E. Brillié, *Engineering*, vol. cxix, April 24, 1925, p. 526.

[14] Ibid., pp. 526–527; also *Organ für die Fortschritte des Eisenbahnwesens*, Heft 2, January 30, 1926, p. 20.

[15] *Engineering*, vol. cxviii, November 21, 1924, pp. 701–703.

[16] Diesel Locomotive with Gear Transmission, *Railway Mechanical Engineer*, vol. 100, no. 7, July, 1926, pp. 425–428.

made of special unburnable material, is removable and can be easily replaced by a new one when its wear exceeds $^3/_{16}$ in. The object of the magnetic operation is to insure a smooth change in speed, and this seems to have been attained during recent tests on the German State Railways.

It is interesting to note the crowded design of the transmission, which hardly permits an increase in power or the addition of a fourth speed.

The locomotive has been already tested on a testing plant and tried out on German Railways, and will soon be shipped to Russia. The results of the actual every-day operation in Russia will be awaited with especial interest.

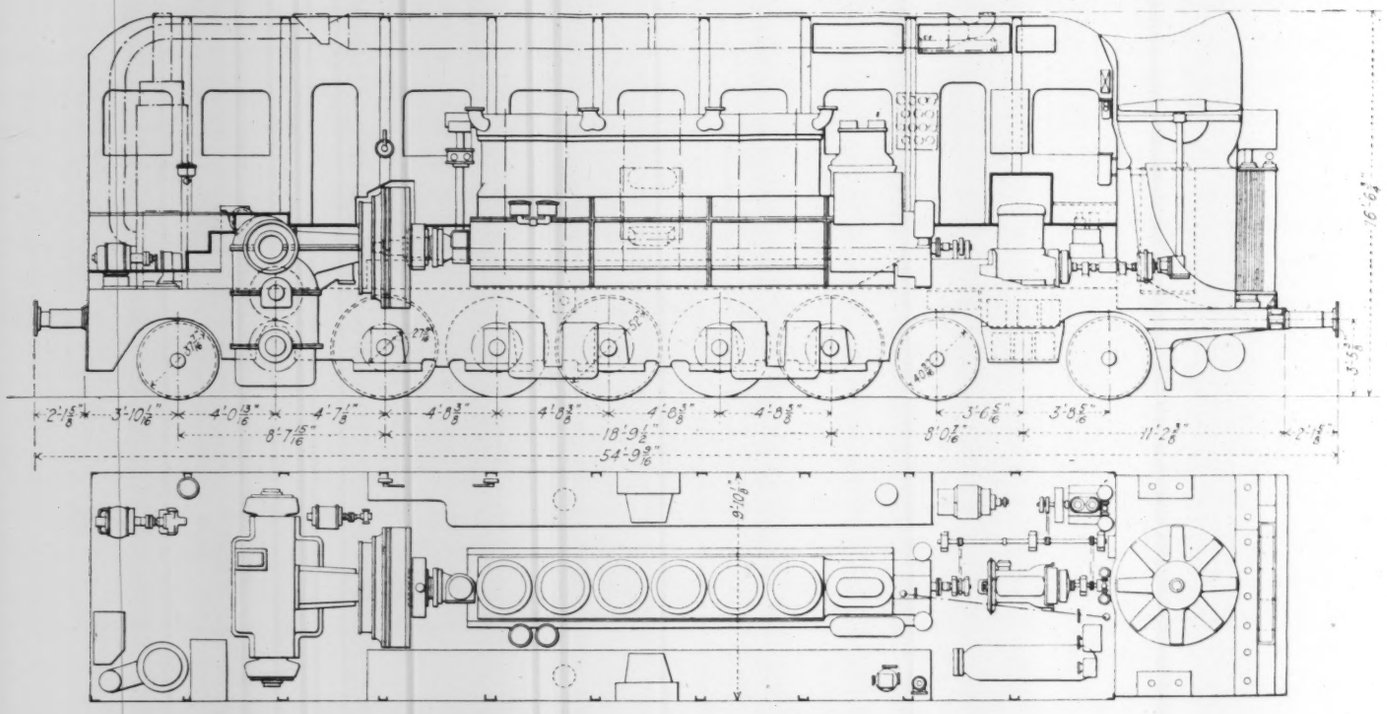

FIG. 26 THE 1000-HP. KRUPP LOCOMOTIVE

2—DIRECT DRIVE

a In 1913, Sulzer Brothers, of Winterthur, Switzerland, built jointly with Borsig Locomotive Works, of Berlin, Germany, a 4–4–4 1000-b.hp. Diesel passenger locomotive with a direct connection between the Diesel-engine crankshaft and the driving wheels by means of side rods. Dr. Diesel himself was the designer of the locomotive, and just like Watt, who did not realize the flexibility and adaptability of the steam engine to locomotive drive, did not seem to be fully cognizant of the inflexibility of the Diesel engine and its poor adaptability to locomotives. He tried to solve the problem by using compressed air for starting the locomotive with the train. The engine had to be run with compressed air until the speed of the train reached the ignition speed, which for this particular locomotive was 6.6 m.p.h. In order to supply enough starting and accelerating power a separate compressed-air outfit had to be provided. The locomotive was therefore equipped with two Diesel engines—a main engine and an auxiliary engine.

The main engine was a 4-cylinder V-type single-acting, two-cycle engine of 1000 b.hp. running at 304 r.p.m. The cylinders were 15 in. in diameter by $21^5/_8$ in. stroke and they all acted on one driving (jack) shaft. The engine was reversible and had arrangements for controlling air and oil admission, and it was thought to obtain in this way a means for changing the mean effective pressure in the cylinders in order to meet variable load conditions. During the tests, indicator cards showed a variation of mean effective pressure from 35 to 170 lb. per sq. in., which means, as compared with ordinary two-cycle engines, a variation of from 40 per cent below to 200 per cent above normal. The auxiliary engine was a vertical two-cylinder, two-cycle engine of 250 hp., cylinders 12 by 15 in., connected with a horizontal multi-stage compressor, the object of this combination being to generate air for starting, supercharging, and other purposes.

Starting by air, however, did not give satisfactory results. In order to obtain good efficiencies it was necessary to apply high rates of expansion, which resulted in low temperatures at the end of expansion and cooled the cylinders to such an extent that ignition became impossible. Short expansions, on the other hand, resulted in very low efficiencies and consequently in high consumptions of air which the auxiliary engine was unable to supply. In addition to this, the driving shaft broke at the end of 1913 and was replaced in the spring of 1914, and some other parts also proved to be defective. The locomotive was all the time undergoing repairs and changes; nevertheless several satisfactory runs were made and valuable experimental data were collected, but the work was abruptly stopped at the beginning of the World War. During the War the locomotive was scrapped.

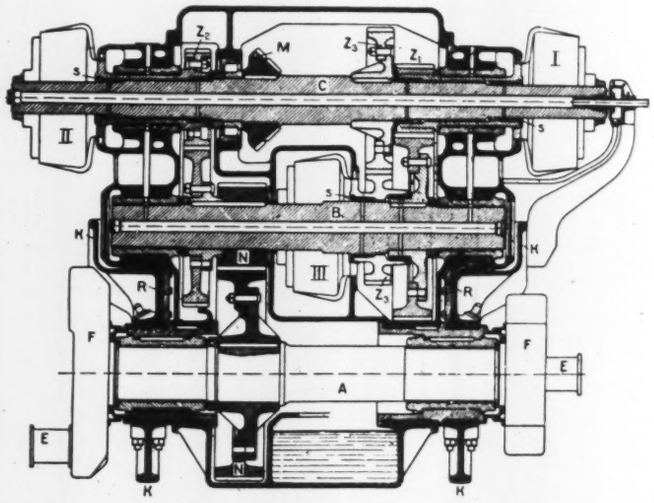

FIG. 27 TRANSMISSION OF 1000-HP. KRUPP LOCOMOTIVE

b The next example of direct connection between the Diesel engine and the driving wheels can be found in the Leroux rail car built in 1913 by the Fives-Lille Co. in France. A 150-b.hp. two-cycle, two-cylinder vertical Diesel engine of the opposed-piston

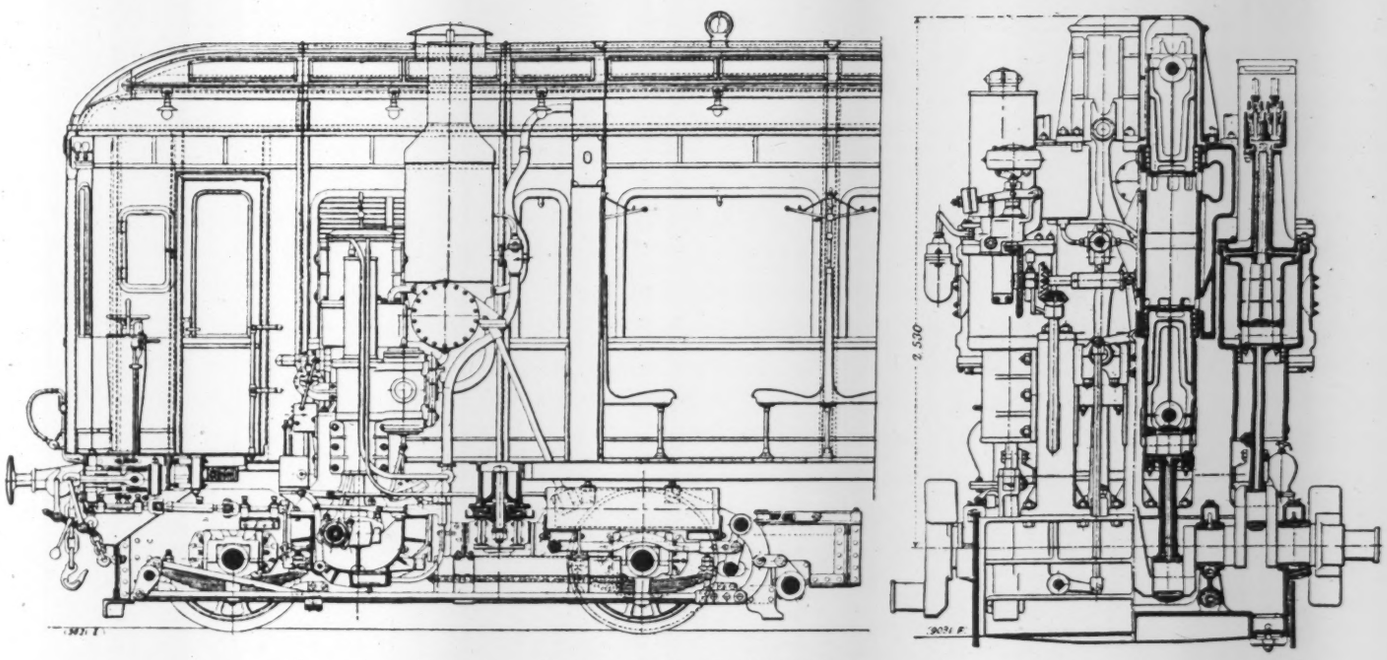

FIG. 28 THE LEROUX RAIL CAR

type with two cranks at 180 deg. was mounted on the front truck (Fig. 28). Two scavenging air pumps were driven from two cranks set on the engine crankshaft at 90 deg. to the oil-engine cranks. The scavenging air-pump cylinders were used for starting the oil engine by compressed air, which latter was provided by a small separately driven compressor and stored in air bottles. Thus the oil-engine cylinders were relieved of the cooling effect of expanding air and the air ignition was not affected as it had been in the case of the Diesel-Sulzer locomotive. In addition, the starting air was preheated by exhaust gases from the oil engine, which resulted in a higher efficiency of the starting arrangement. The car kept running until the fall of 1914, when, unfortunately, it was destroyed during the German occupation of Northern France. The weight of the car was 25.8 tons.[17] With a train weighing 68.2

[17] Traction on Rails by Internal Combustion Engines, by M. E. Brillié, *Engineering*, vol. 119, May 29, 1925, pp. 684–685.

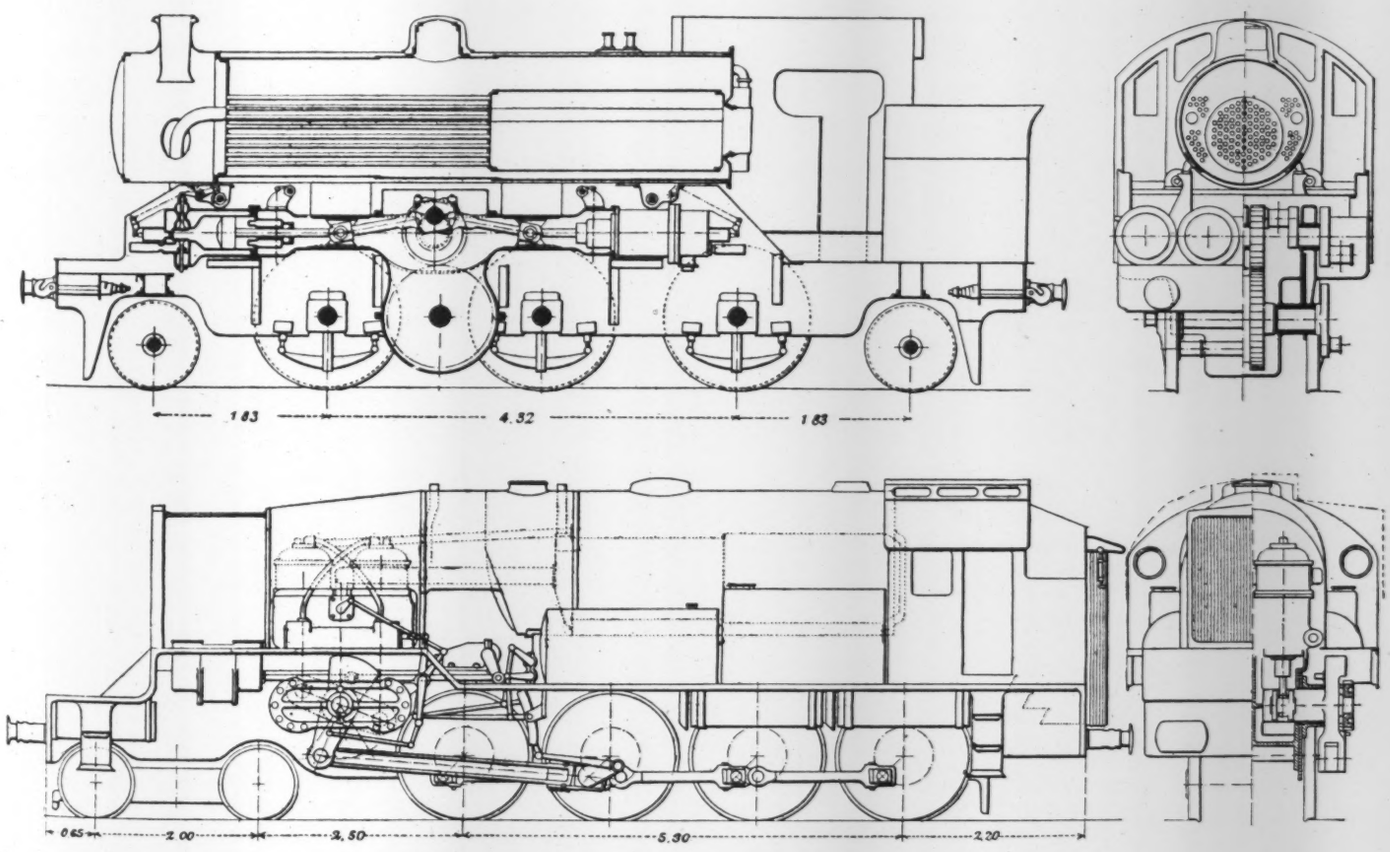

FIG. 29 THE KITSON-STILL LOCOMOTIVE (*Above*)

FIG. 30 THE SCHNEIDER-STILL LOCOMOTIVE (*Below*)

tons (motor car included) the fuel consumption was 1.74 lb. of oil per mile, or 2.56 lb. per 100 ton-miles.

c Two direct-driven locomotives of considerable size, embodying the Still principles of starting, are at present under construction; one is being built in Leeds, England, at the plant of Messrs. Kitson & Co., and another in France, at the works of Schneider & Co. in Creusot. The Still engine is a combination of an oil engine with a steam engine, steam being generated in a separate boiler by the waste heat of the oil engine, and being expanded in the oil-engine cylinder on the other side of the piston. The boiler is placed in communication with the water jackets of the engine cylinders, and thus both the cylinder-jacket heat and the exhaust-gas heat are utilized for steam generated in the Still engine.

Several Still engines have been built for marine propulsion, the latest installation being that on the motorship *Dolius*. It consists of two four-cylinder, two-cycle, single-acting engines, each developing 1250 b.hp. at 120 r.p.m. Official tests made by the Marine Oil-Engine Trials Committee showed fuel consumptions of 0.353 and 0.358 lb. of oil per b.hp-hr. at full load and three-quarters load, corresponding to thermal efficiencies of 37.1 and 36.4 per cent,[18] respectively.

The Still engine, in addition to its low fuel consumption, offers a very good combination for train starting, steam being an ideal fluid for this purpose. Of course, if the engine has not been run before starting, it becomes necessary to heat the boiler with a specially provided oil burner in order to generate steam for starting. However, in so doing the cylinders are warmed up, ignition is facilitated, and starting on oil is rendered much easier. The presence of a boiler on an oil-engine locomotive may prove to be of great value, as it offers a large amount of stored and available energy, especially with the oil burner in operation, which can be utilized on heavy grades and in emergencies, even though at the sacrifice of fuel economy. There are of course disadvantages to the system, which will be given below.

The Kitson-Still locomotive is shown in Fig. 29. The oil engine, rated at 1000 b.hp., is of the four-cycle type and has eight horizontal cylinders driving a four-crank shaft geared to a jackshaft placed underneath the boiler. The latter is of the locomotive type with a modified firebox. The outside cylinder ends are those of the oil engine, the inside ends being those of the steam engine. Starting takes place through the action of steam on the inside surfaces of all eight pistons. The estimated weight of the locomotive is 77 tons, i.e., 154 lb. per b.hp., or almost the same as in ordinary steam locomotives. The wheel arrangement is 2–6–2 and the total length about 37 ft.

The Schneider-Still locomotive is shown in Fig. 30. It is of the 4–8–0 type, has an overall length about 45 ft. with a rigid wheel-base of 17 ft. 4½ in., and the diameter of the driving wheels is 5 ft. 2 in. The oil engine was designed as a two-cycle, four-cylinder, vertical engine developing 1250 b.hp. at 300 r.p.m. The pistons act on three-arm rocking levers connected to ordinary main rods. The boiler is similar to that of the Kitson engine. As the engine is of the two-cycle type and has comparatively smaller pistons, starting of the locomotive is supposed to be effected by using steam on both sides of the pistons. The tractive-effort curves are shown in Fig. 31. *A* is the curve resulting from steam action only, when the oil burner is applied. *B* is the curve of the Still engine (both oil engine and action of steam from waste heat). *C* is the combined tractive effort for overload and emergency conditions, part *D* corresponding to steam action on both sides of pistons. Curves *a*, *b* and *c* represent corresponding power outputs. The estimated weight on the drivers is 141,000 lb. The starting tractive effort would thus give a factor of adhesion of 4.64. The total weight of the locomotive will be in the neighborhood of 100 tons, or 150 lb. per b.hp.[19]

Information received from abroad is to the effect that the oil-engine part of the locomotive has been recently redesigned, the four vertical cylinders having been replaced by six horizontal cylinders. Thus the design of the locomotive comes closer to that of the Kitson locomotive.

[18] Second Report of the Marine Oil-Engine Trials Committee, read before The Institution of Mechanical Engineers and The Institution of Naval Architects, London, March 20, 1925, pp. 54–57.

[19] *Engineering*, vol. cxix, May 29, 1925, pp. 685–687.

The two Still locomotives represent a radical departure from previous oil-engine locomotives. A locomotive with a constant-power prime mover and a transmission of this kind described under Class A, B, or C-1 has a hyperbolic speed-tractive-effort characteristic which drops very rapidly with the increase of speed. The locomotives with direct drive (C-2), if at all possible, give a straight-line (constant torque) characteristic with an increase at very low speeds for starting. The steam-locomotive speed-tractive-effort curve lies somewhere between the hyperbola and straight line, and the Still locomotive's curve comes closer to the steam line than that of any other oil-engine locomotive. This is of course its advantage, resulting as do many other advantages from the fact that the Still locomotive is a combination of an oil engine with a steam engine. This combination, however, especially in a locomotive, has its disadvantages; for instance, the realization of tractive efforts beyond certain limits will require the application of the oil burner and will cause a loss in thermal efficiency. The boiler will then be subjected to wear, accumulation of scale, stand-by losses, and maintenance repairs, although to a lesser degree than in ordinary steam locomotives, but nevertheless to an extent depending upon the length of time during which curve *C* (Fig. 31) is used instead of *B*. If steam is very often used at low speeds on heavy grades it may impose on the boiler a great amount of work and result in an appreciable increase in fuel consumption and cost of maintenance. The necessity of using oil for the burner in order to be able to control the fire easily, may not be favored in places where coal is cheap. On the other hand, on level roads, with uniform conditions of traffic, the Still locomotive, especially with passenger through trains, may

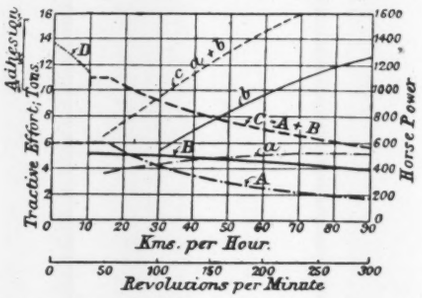

FIG. 31 TRACTIVE-EFFORT CURVES, SCHNEIDER-STILL LOCOMOTIVE (From *Engineering*.)

run for long periods of time on oil with very little steam, and that probably steam which is being generated by waste heat from the oil-engine exhaust.

Further, three other advantages of the oil-engine locomotive proper disappear: the Still locomotive ceases to be smokeless, it is not always available for immediate service, and it requires a certain supply of water. It is true, however, that the latter is partly offset by the possibility of elimination of cooling arrangements, which are not needed on a Still locomotive.

All these points will undoubtedly be cleared up as soon as the locomotives are placed in service, and they will be watched with unabating interest by locomotive and railroad men.

The author's classification of mechanical transmissions would not be complete if he did not mention an ingenious variable transmission called the Constantinesco torque converter. The theory of this transmission is extremely involved and will not be discussed here. It was fully expounded in a series of articles published by G. Constantinesco in the *Automobile Engineer*, London, issues of November and December, 1923, and of February and March, 1924. It represents a mechanism with two degrees of freedom, one of which is taken up by the inertia of an oscillating mass. It therefore comes under the class of differential transmissions, and with it an automatic variation of speed and torque can be obtained.

This gear was applied to an automobile and also to a 250-hp. Diesel locomotive exhibited at the Empire Exhibition in London. No particulars regarding the performance or service of the locomotive are available at present.[20]

[20] See also paper entitled Transmission of Power—The Present, The Future, read by G. Constantinesco before the North-East Coast Institution of Engineers and Shipbuilders.

An Analysis of Deep-Well Oil Pumping

By W. HOWARD CLAPP,[1] PASADENA, CAL.

The paper points out that present methods of pumping oil from underground are unsatisfactory. Oil is brought to the surface by plunger pumps with sucker-rod connections to the familiar walking beam, pitman, wristpin, and bandwheel. The inadequacy of the method has led to the invention of innumerable devices for obtaining smoother operation which have mostly been discarded after failure. The present mechanisms are analyzed and shown to be far from ideal. The author then develops an ideal motion where abrupt forces are reduced to a minimum, thus reducing fatigue stresses and other causes of failure. The ideal kinematic conditions are difficult to realize by applying mechanism to the ordinary type of prime mover. The author develops the design of an air-driven engine which should have an indicator card corresponding to ideal kinematic requirements, and summarizes its advantages in reducing chances of breakage in sucker rods and reducing friction in the actuating mechanism. The combination of a central compressor plant actuated by internal combustion engines is shown to give the possibility of a cheaper source of power.

THERE are no engineers in the oil industry who are satisfied with present methods of pumping oil from underground. Some experimenting is being carried on with the air lift and with a compressed-gas lift; the latter in some cases with considerable success, in others, for reasons unknown, with complete failure. Recently, some of the larger companies have been conducting tests on the oil-pumping mechanism to find out what is actually taking place. Most of the oil is brought to the surface by plunger pumps with sucker-rod connections to the familiar walking beam, pitman, wristpin, and bandwheel. The inadequacy of the actuating mechanism used is everywhere conceded, and innumerable substitutes have been invented with the object of obtaining smoother operation, longer strokes, and straight strokes. The fate of these devices may be seen by any one who visits a producing field, where such scrapped equipment, tried and discarded, serves to emphasize a need which has not yet been met. Meanwhile, wells are being drilled to increased depths; few of them are straight, and water and sand increase the difficulty of pumping. The need for a better pumping device is a challenge to the engineering profession.

The average depth of the wells in the Los Angeles fields is between 4000 and 5000 ft. Some are 7000 ft. deep. The stress in the upper part of the rod due to its weight is about 13,000 lb. per sq. in. for 4000 ft. of rod, but this is frequently quadrupled because of the weight of the oil column, frictional resistance, and inertia. Under the varying load the rods change length perhaps 15 in. each cycle. Because of this varying stretch the bottom of the rod may be going down while the top is rising. Again the weight of the column of oil is abruptly applied. Frequently the valves close sharply after some initial velocity of the plunger has been reached. This loading sets up elastic waves which travel through the steel with a speed of about 12,000 ft. per sec. Broken and unscrewed rods are frequent and expensive occurrences. An exact analysis of the stresses set up under these conditions is probably impossible. There will be some type of motion and time of cycle which for any particular length of rod will be most satisfactory. In order to make an analysis, the rod is here assumed to move like a rigid body and the assumption is made that a type of motion which would be least severe for this case will also be best for the elastic rod.

Forces Acting on the Pump Rods

Fig. 1 shows the manner in which the forces at the upper end of the sucker rods vary during the two strokes of a pump cycle. Four forces make up the resultant force P which stresses the rod at any instant. These forces are designated as follows:

W_r = weight of the sucker rods and plunger mechanism
W_o = weight of the column of oil that is being lifted
F_1 = frictional resistances on the upstroke
F_2 = frictional resistances on the downstroke
I = inertia forces whose value depends upon the weight that is being lifted and the character of the motion given to the plunger.

1 At the beginning of the upstroke the force acting upon the upper end of the sucker rod is

$$P_1 = W_r + W_o + F_1 + I_1.$$

2 At the end of the upstroke

$$P_2 = W_r + W_o + F_1 - I_2.$$

3 At the beginning the downstroke

$$P_3 = W_r - F_2 - I_3.$$

4 At the end of the downstroke

$$P_4 = W_r - F_2 + I_4.$$

5 At any position during a stroke the force P acting upon the rod is given by the ordinate of the unshaded portion of the diagram, as at m on the upstroke or n on the downstroke.

The forces W_r and W_o are constant in amount at any particular

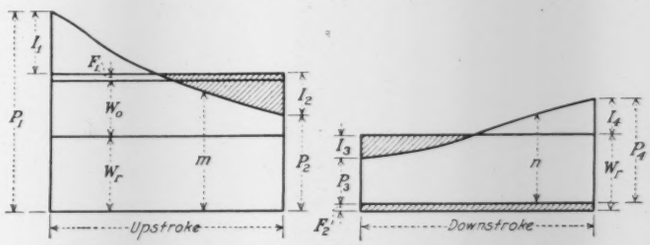

Fig. 1 Variation of Forces at the Upper End of the Sucker Rods During the Two Strokes of a Pump Cycle

well. The diagram, Fig. 1, shows W_r as 4000 lb. and W_o as 3000 lb. The frictional forces F_1 and F_2 are not measurable with great exactness, but are of about the order shown except where the well is crooked or is "making sand." F_1, the frictional resistance to the motion of the sucker rods, the plunger, and the column of oil on the upstroke, *adds* to the total force P on the rod. F_2, the frictional resistance on the downstroke, is the resistance to the motion of the plunger and the rods, but not to that of the column of oil, since the latter does not return with the plunger. These forces are assumed to have a constant value throughout a stroke.

Inertia Forces

The inertia force I is a variable force whose value depends upon the weight being raised and the character of the motion that is imparted to the sucker-rod chain. In Fig. 1 this force is shown in correct proportion to the others for the standard walking-beam mechanism when making a 48-in. stroke at the rate of 24 strokes per minute. The motion of a pump rod so actuated is identical in character with that of an engine crosshead making the same number of 48-in. strokes and having a ratio of connecting rod to crank identical with the ratio of pitman length to wristpin-arm length. The amount of this force in pounds is given by the familiar equation:

$$I = \frac{WCN^2}{35{,}200}\left(\cos\theta + \cos\frac{2\theta}{n}\right),$$

where I = inertia force any instant, lb.
C = wristpin-arm length, in.
N = r.p.m. of bandwheel
W = weight that is being accelerated, lb.
θ = angle through which the wristpin has moved, measured from its uppermost position, deg.
n = ratio of pitman length to wristpin-arm length.

[1] Professor of Machine Design, California Institute of Technology. Mem. A.S.M.E.

Presented at the January, 1926, meeting of the Los Angeles Section of the A.S.M.E.

Applying this equation to the particular example referred to in Fig. 1, it is found that

$$P_1 = 4000 + 3000 + 3210 + F_1 = 10{,}210 + F_1 \text{ lb.}$$
$$P_2 = 4000 + 3000 - 2290 + F_1 = 4710 + F_1 \text{ lb.}$$
$$P_3 = 4000 - 1310 - F_2 = 2690 - F_2 \text{ lb.}$$
$$P_4 = 4000 + 1833 - F_2 = 5833 - F_2 \text{ lb.}$$

Fatigue Failure

Under the stress produced by the variable force P, which in this particular case is shown to vary from a maximum of 10,210 lb. plus the frictional resistance to a minimum of 2690 lb. minus the frictional resistance and to have an abrupt change of the load at the end of each stroke, breakage of sucker rods is a frequent occurrence and the expense due to delays of pumping and the cost of repairs is very high. The type of break under such varying stresses is commonly referred to as "fatigue failure," but the term is a rather unfortunate one, since the metal does not grow tired and quit, nor does it undergo chemical change as in the case of "lead sickness." The nature of this failure is now well understood, and a reference to it is necessary to the analysis that is here undertaken. It is more scientifically referred to as "progressive fracture" of the

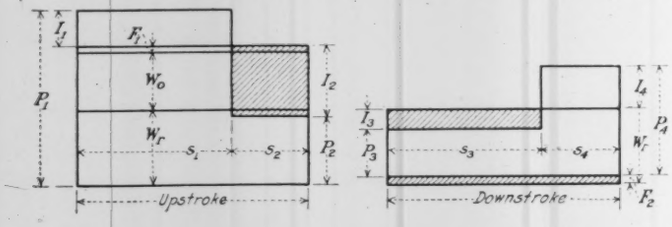

FIG. 2 DIAGRAM OF FORCES ACTING UPON THE ROD WHEN ACCELERATION HAS BEEN CARRIED SOMEWHAT BEYOND MID-STROKE

steel, and takes place as follows: Under the stress of a varying load, some part of the rod (where there is a kink or a scratch on the metal surface, which brings about a localized stress of high intensity) will start to fail. This failure will consist of a fracture of one of the minute crystals of the metal along one of its cleavage planes. In turn, the fracture sets up in the next adjacent crystal a stress of high intensity, and eventually this crystal also is fractured. Thus, under the prying action of the variable load, the fracture extends gradually around the rod and into the rod until the reduced section, no longer sufficient to carry the load, gives way.

Relation of the Failure to the Character of the Load

It has been demonstrated by Wöhler,[2] Bauschinger,[3] and Weyrauch,[4] and more recently and thoroughly by Moore and Kommers[5] that this type of failure is influenced by four different factors. First, failure depends upon the maximum stress: the higher the maximum stress, the sooner failure will occur. Second, failure is a function of the range of stress: the greater the range between maximum and minimum stress, the sooner failure will occur. Third, failure increases with the abruptness with which the load is applied and released; it is demonstrable that a load instantly applied will produce twice as high a stress as would occur if the load were gradually applied. Fourth, failure depends upon the number of repetitions of the variable loading.

Objectionable Features of the Force Curve

Applying these criteria of failure to the diagram of Fig. 1, we see that failure increases as P_1 is increased; second, that failure increases as the difference between P_1 and P_3 (the least force) increases; and, finally, that the failure will be hastened because of the abrupt changes of load which occur at the end of each stroke, from P_4 to P_1 and from P_2 to P_3, respectively. To illustrate with numerical values: the maximum intensity of stress is equal to the maximum load (10,210 + F_1 lb.) divided by the cross-sectional area of the sucker rod. Similarly, the maximum range of stress is (7520 + F_1 + F_2) divided by the rod area, while the instantaneous change of load from P_1 to P_4 is (4377 + F_1 + F_2) lb., and that from F_2 to P_3 is (2020 + F_1 + F_2) lb.

Conditions Favoring Least Maximum and Least Range of Stress

Of the four forces which make up the variable force P there is only one which may be modified in order to obtain more nearly ideal stress conditions. This is the inertia force, I, whose value can be changed by modifying the character of the motion imparted to the weights W_r and W_o. We shall now determine what modification of the stroke motion will bring about the best possible stress conditions.

Considering, for the present, only the first two criteria of failure and referring to Fig. 1, it will be seen that P_1 can be reduced by the same amount by which it is possible to reduce I_1. The inertia force I varies directly with the acceleration—in fact, the curve of inertia force may represent a curve of acceleration drawn to proper scale. It is apparent that if a weight W is to be raised through a distance S in a time T, the acceleration will be a minimum if it be maintained at a constant value throughout the interval. In this case the body would not be brought to rest, but would be traveling at its highest speed at the end of the stroke. If the acceleration is maintained constant to mid-stroke, an equal constant deceleration will be required to bring the body to rest at the end of the stroke. The acceleration will then be twice as great as in the first case, if the body is to complete the stroke in the same time. Fig. 2 is a diagram of the forces acting upon the rod for a case in which the acceleration has been carried somewhat beyond mid-stroke. The same weights and time of lifting are assumed as in Fig. 1. By carrying the acceleration beyond midstroke the inertia force I_1 is reduced materially while I_2 is increased in amount. If the return stroke be made in the same time and manner, I_3 will bear the same relation to I_1 as the weight W_r being lowered does to the weights (W_r + W_o) being raised, which, for the particular case we have been considering, makes I_3 equal to four-sevenths of I_1. Similarly, I_4 equals four-sevenths of I_2. The result of this change in the character of the motion is to decrease both the maximum load P_1 and also the range of load $(P_1 - P_3)$, and it is apparent that this improvement may be still further promoted by using an even smaller acceleration on the upstroke and continuing for an even longer proportion of the stroke, s_1, carrying this out until the final force P_2 becomes equal to P_3. To go further is to make P_2 less than P_3, but this will increase the range of stress, which is dependent upon the difference between maximum and minimum loads. An additional improvement may, however, be made by taking a longer proportion of the time of the cycle for the upstroke than for the downstroke. This will decrease P_1 still further, but at the expense of P_3. Now if while making this change we continue to keep P_2 equal to P_3 by still further increasing the ratio of s_1 to s_2 (equal to ratio of s_3 to s_4) the effect is to increase P_4 as P_1 decreases, and such increase can be carried on advantageously until P_4 becomes equal to P_1. Now, these are the relationships which should exist in order to have the least maximum load and least range of load on the sucker rod: namely, $P_1 = P_4$ and $P_2 = P_3$. We shall now determine how great an improvement may be effected by employing this ideal motion.

Relations Between the Accelerations, Spaces Covered, and Time Periods

From the equations given previously,

$$W_r + W_o + F_1 + I_1 = W_r - F_2 + I_4 \quad [1]$$

or

$$I_4 - I_1 = W_o + F_1 + F_2$$

Similarly, since $P_2 = P_3$

$$W_r + W_o + F_1 - I_2 = W_r - F_2 - I_3$$

or

$$I_2 - I_3 = W_o + F_1 + F_2 \quad [1']$$

Therefore

$$I_1 + I_2 = I_3 + I_4 \quad [2]$$

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[2] Ueber die Festigkeitsversuche mit Eisen und Stahl, 1870.

[3] Mittheilungen der Konig, J. Bauschinger, Münich, 1886.

[4] Structures of Iron and Steel, J. Weyrauch, translated by A. J. Du Bois, 1877.

[5] *Iron Age*, vol. 105, no. 23; also Am. Soc. for Testing Materials, 1926, et seq.

and

$$I_1 = I_3 \text{ and } I_2 = I_4$$

Let a_1 = acceleration which acts during first part of upstroke
s_1 = distance through which weights W_r and W_o move with this acceleration
t_1 = time required for the above action
$a_2; s_2; t_2$ = corresponding quantities for retardation period of upstroke
$a_3; s_3; t_3$ = corresponding quantities for acceleration period of downstroke
$a_4; s_4; t_4$ = corresponding quantities for retardation period of downstroke
T_1 = time for upstroke $= t_1 + t_2$
T_2 = time for return stroke $= t_3 + t_4$
A_1 = acceleration which would cause the motion over the stroke S in the time T_1 when acting constantly throughout the stroke
g = acceleration of gravity expressed in the same units as the other acceleration.

From Equations [1] and [2] and the laws for uniform acceleration there result two very important relationships which may be stated as follows:

(*a*) The proper ratio of the time of the return stroke to the time of the working stroke is

$$\frac{T_2}{T_1} = \sqrt{\frac{W_r}{W_r + W_o}} \quad \ldots\ldots [3][6]$$

(*b*) The space s_1, covered during the accelerating period ranges from 80 per cent to 90 per cent of the entire stroke. It becomes greater with longer strokes and with slower speeds. The relationship is rather complex, but is expressed by the following equation:

$$\frac{S}{s_1} = 1 - \frac{KT_1^2}{4S} + \sqrt{1 + \frac{(KT_1^2)^2}{4S}} \quad \ldots\ldots [4][6]$$

where K is a constant with the value $(W_0 + F_1 + F_2)g/(W_r + W_0)$.

Application of Mathematical Relationships to a Typical Case

Taking the previous illustrative example: $W_r = 4000$ lb.; $W_0 = 3000$ lb.; $S = 4$ ft.; $N = 24$ r.p.m. Assume also that $F_1 = 350$ lb. and $F_2 = 240$ lb. Then:

$$\frac{T_2}{T_1} = \sqrt{\frac{W_r}{W_r + W_o}} = \sqrt{\frac{4000}{7000}} = 0.7559$$

$$T = T_1 + T_2 = \frac{60}{24} = 2.5 \text{ sec.}$$

$$T_1 = \frac{2.5}{1.7559} = 1.4238 \text{ sec.}$$

$$T_2 = (2.5 - 1.4238) = 1.0762 \text{ sec.}$$

$$K = \frac{(3000 + 350 + 240) \times 32.2}{4000 + 3000} = 16.514$$

$$\frac{KT_1^2}{4S} = 2.0923 \qquad \left(\frac{KT_1^2}{4S}\right)^2 = 4.3778$$

$$\sqrt{1 + \frac{(KT_1^2)^2}{4S}} = \mathbf{2.319}$$

$$\frac{S}{s_1} = 1 - 2.0923 + 2.319 = \mathbf{1.2267}$$

$$A_1 = \frac{2S}{T_1^2} = 3.9463 \text{ ft. per sec. per sec.}$$

$$a_1 = \frac{A_1 S}{s_1} = 4.841 \text{ ft. per sec. per sec.}$$

$$a_2 = (a_1 + K) = 23.355 \text{ ft. per sec. per sec.}$$

[6] The analytical derivations of [3] and [4] are given in an appendix to the paper not included in this publication.

$$s_1 = \frac{48}{1.2267} = 39.13 \text{ in.}$$

$$t_1 = \frac{1.4239}{1.2267} = 1.1606 \text{ sec.}$$

(To check: $s_1 = {}^1/_2\, a_1 t_1^2 = 3.2608$ ft. $= 39.1296$ in.)

$$I_1 = \left(\frac{W_r + W_o}{g}\right) a_1 = 1052.4 \text{ lb.} = I_3$$

$$I_2 = \left(\frac{W_r + W_o}{g}\right) a_2 = 4642.4 \text{ lb.} = I_4$$

We now have

$$P_1 = 7000 + 350 + \mathbf{1052.4} = \mathbf{8402.4} \text{ lb.}$$
$$P_4 = 4000 - 240 + \mathbf{4642.4} = \mathbf{8402.4} \text{ lb.}$$
$$P_2 = 7000 + 350 - \mathbf{4642.4} = \mathbf{2707.6} \text{ lb.}$$
$$P_3 = 4000 - 240 - \mathbf{1052.4} = \mathbf{2707.6} \text{ lb.}$$

Advantages Gained

If the pump rod of the well considered in these examples could be given the type of motion determined by these relationships, the

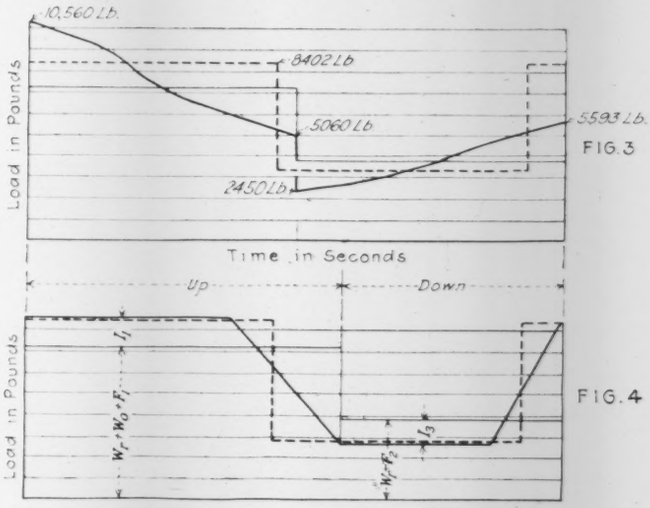

FIG. 3

FIG. 4

Fig. 3 Force Variation for Ordinary and Ideal Types of Motion (Ideal Force Curve Shown in Dotted Lines)

Fig. 4 Force Curve Where Change from Maximum to Minimum Load Occurs Less Abruptly

advantages gained through a reduction in the maximum load and the range of load would be most substantial. Comparing the results obtained with those of the walking-beam type of motion and using the same value for friction in each case, we have

	Max. load on rod	Max. range of load
Walking-beam mechanism	10,560 lb.	8,110 lb.
Ideal motion	8,402 lb.	5,695 lb.
Reduction in maximum load	2,158 lb. or 20.5 per cent	
Reduction in range of load		2,415 lb. or 29.7 per cent

The force variation for the two types of motion is shown graphically in Fig. 3, the ideal force curve being shown in dotted lines.

A motion of the character which has thus been analytically determined is ideal in that it is not possible to raise a given load over a given distance in a given time with any lower maximum force or range of force than is thus obtained. Since "fatigue failure" has been definitely shown to be influenced by both of these factors, a lowering of these values as shown above will contribute greatly toward reducing the number of such failures.

Applying the Third Criterion of Fatigue Failure by Eliminating Abrupt Changes of Force

While the force curve shown by the dotted lines of Fig. 3 gives the least maximum and least range of load possible for the given stroke and time period, it violates the third criterion of "fatigue failure," since the changes from maximum to minimum load occur

abruptly. It is probably better to remedy this as shown by the full-line curve of Fig. 4. Calculation shows that an increase in the maximum load of 35 lb., or less than one-half of one per cent, will increase the velocity sufficiently to permit of a gradual reduction from maximum to minimum load, as shown, distributed over about 38 per cent of the stroke. This compromise would no doubt be warranted in the results obtained, although we are not yet in possession of sufficient data to tell much about the comparative influence of these factors.

Developing a Practical Mechanism

The first object of this paper has now been accomplished, and we have determined by analysis a type of motion which will keep the breakage of sucker rods at a minimum. The second part, that of designing a mechanism which will bring about such a motion, partakes of the nature of invention. Any mechanism that might be devised should also meet a number of practical requirements. (1) The mechanism should be simple, with few moving parts, and the mechanical efficiency should be at least as high as that of the present drive. (2) It should be foolproof. (3) Provision should be made for a stroke of varying length with a maximum of perhaps eight feet, and it should be possible to increase or decrease the rate of pumping at will to meet the requirements at any particular well. Finally, (4) the mechanism should stop at once if the well starts to "sand up," thus avoiding any breakage of sucker rods. Such

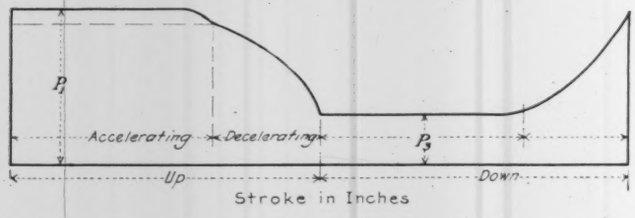

FIG. 5 FORCES OF FIG. 4 REPLOTTED ON DISPLACEMENT AS ABSCISSA

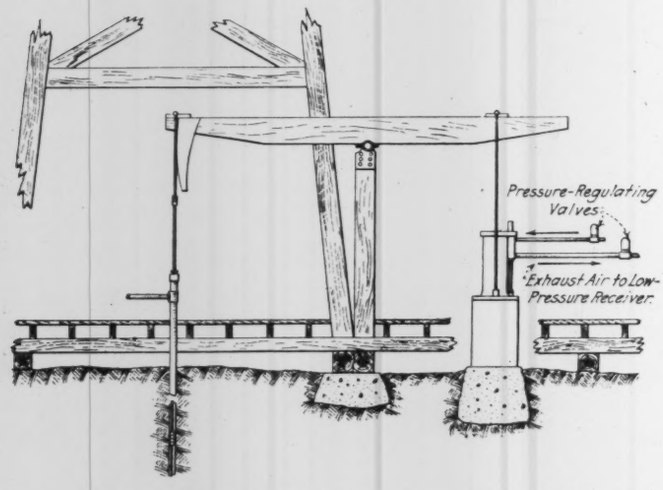

FIG. 6 IDEAL AIR-ACTUATED ENGINE FOR DRIVE OF DEEP-WELL PUMP; GENERAL ARRANGEMENT

a combination of demands would seem to be insurmountable; certainly it would discourage any one from trying to invent any combination of links, or of linkage and sliding members, or of cam-actuated parts to meet such exacting conditions. However, as with many apparently tough problems, there is one method of solution which when developed meets all of the requirements so admirably and, in addition, offers other attractive features so obvious that one wonders why it was not used years ago.

The Ideal Force Curve an Indicator Card

If the modified force curve of Fig. 4 be replotted to coördinates of force and displacement instead of force and time we have the curve of Fig. 5, which is an engine indicator card. The indicator card has several peculiar features: (1) the back pressure P_3 is about three-eighths of the forward pressure P_1, (2) compression pressure is carried up to the pressure at admission, and (3) the cut-off occurs as late in the stroke as possible to permit of expansion down to the back pressure P_3. It will be seen that no engine ever had an expansion curve like the one shown in Fig. 5, but the essential thing is that the change from maximum to minimum pressure be made in a gradual manner. Therefore, if an engine can be designed to operate without a flywheel and develop a length of stroke which may be varied at will, and if such an engine be made to operate the sucker rods either by direct attachment or through the walking beam as shown in Fig. 6, it will automatically bring about a motion that will have the required characteristics.

An Engine Designed to Meet Ideal Conditions

The engine shown in Figs. 6, 7, 8, and 9 is designed to run with

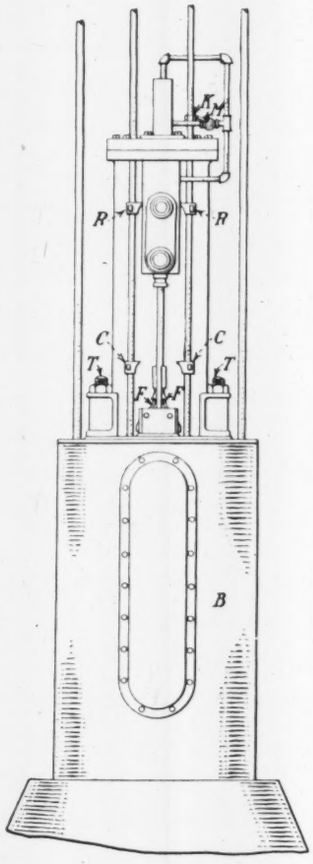

FIG. 7

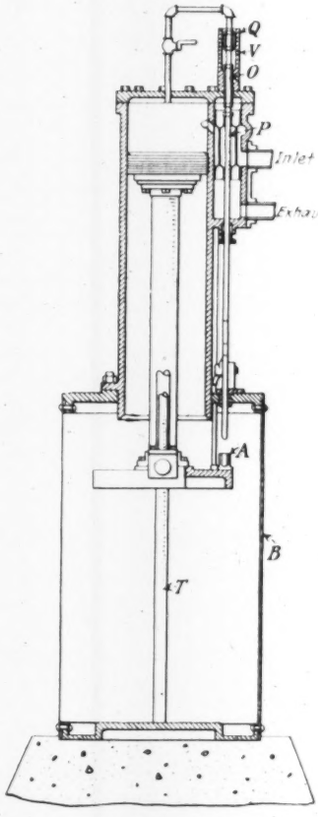

FIG. 8.

FIG. 7 ENGINE FOUNDATION, TIE RODS, AND GUIDES FOR CROSSHEAD

FIG. 8 VALVE IN UPPERMOST POSITION

air under a working pressure of from 80 to 100 lb. per sq. in. gage and with the exhaust pressure as required to bring about the ideal motion. The distinctive feature is that the air moves in a closed circuit, the exhaust air going to a low-pressure receiver, whence it is compressed and sent to a high-pressure receiver for use over again. The engine cylinder is set upon a light steel base B, and is held to the foundation by two tie rods T, T, which also serve as the guides for the crosshead H. The walking beam is actuated by two connecting links L, L which are attached to the crosshead as shown in Fig. 9. The arrangement is one that favors lightness and simplicity, for the cylinder and base B are tension members, while the piston rod is a compression member. No stuffing box is required. The valve mechanism operates as follows: the piston valve P has inside admission and is sufficiently unbalanced (the lower diameter being slightly larger than the upper) so that it tends to move downward. In Figs. 8, 10, and 11 the valve is shown in its uppermost or admission position. It is held in this position by the two pivoted fingers F, F until the cut-off tappets C, C, which move up and down with the crosshead, pass by and engage the fingers for an instant. This releases the valve for a 2-in. drop, after which it is again brought to rest by the fingers. The valve

now closes the port opening. The air expands in the cylinder until the release tappets R, R engage the fingers, permitting the valve to drop again to the exhaust position. These release tappets are set to act when the piston has almost reached the limit of its downward travel. The piston and crosshead now start to return since the back pressure is less than the force exerted by the descending rods. The velocity increases under this accelerating force until it is checked by the compression. Compression is brought about as follows: The piston valve is in its lowest position during exhaust, but as the crosshead moves upward the compression tappet K engages and opens a valve M, (Figs. 7 and 10) which allows air at working pressure to enter behind the auxiliary piston Q. This reverses the direction of unbalance of the piston valve, and it moves

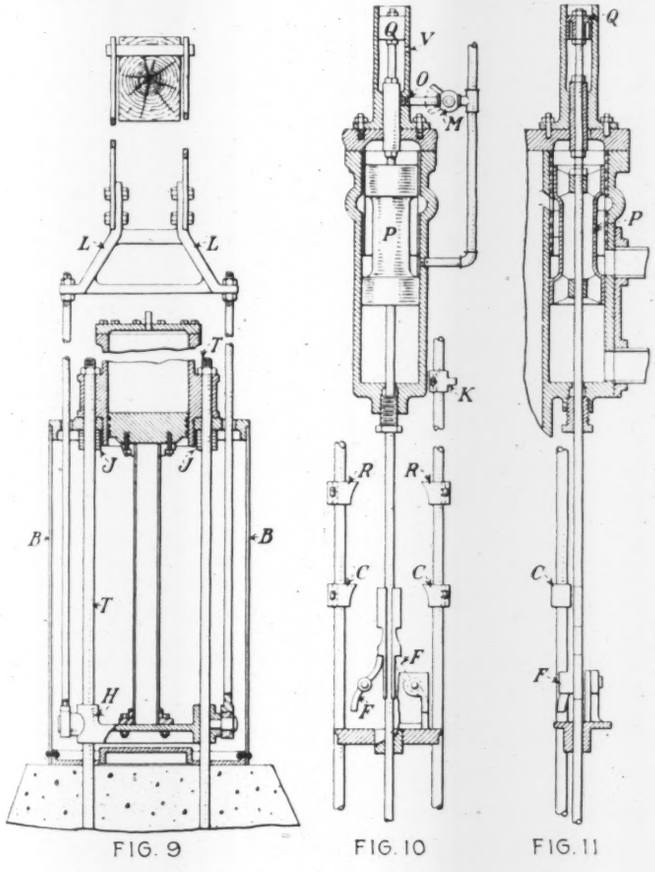

FIG. 9 WALKING BEAM ACTUATED BY LINKS L, L ATTACHED TO CROSSHEAD

FIG. 10 VALVE MECHANISM FIG. 11 VALVE MECHANISM

upward 2 in. but is stopped by the shutting off of the opening O by which the air entered the auxiliary piston, and by the almost simultaneous opening of the vent V which exhausts the air behind Q to the atmosphere. The piston valve is now held on the fingers F, F in the same position as at cut-off. When the compression tappet K is properly set and the pressures correctly adjusted to the load and desired cycle, the piston will come to rest just before it would close the port opening. If for any reason the compression tappet is set late, the piston will close the port opening and compression will increase behind the piston to a degree which will bring the piston to an almost immediate stop. The final lift of the piston valve is effected by the admission tappet A, which is carried on the crosshead and makes a dashpot engagement with the end of the valve rod.

REGULATION OF THE LENGTH OF STROKE AND THE NUMBER OF STROKES PER MINUTE

There is a proper admission and exhaust pressure and a proper setting of the cut-off, release, and compression tappets for any desired length of stroke or rate of cycle at any given well. The indicator card should show the following characteristics if the valve setting is correct: (*a*) Compression up to admission pressure, with stroke ending as soon as this is effected; (*b*) expansion down to, or nearly to, the back pressure; (*c*) a proper relation between the working pressure and the back pressure such that the time of the return stroke is to the time of the working stroke as $\sqrt{W_r/(W_r + W_o)}$. This is the relationship established in the first part of this paper as the ideal time rate. It is entirely independent of the friction within the well and may be predetermined for any well with fair exactness. It is to be noted that only a small increase in pressure is necessary to bring about a considerable increase in the speed of pumping. Thus in Fig. 12, the total pressure behind the piston until cut-off is P_1, but it is only the *excess* pressure I_1 which gives increase of speed to the piston. This force starts to diminish at cut-off C, and changes to a resistance after point D. The greatest velocity of the piston is at D, at which time the kinetic energy in the reciprocating parts is represented by the area $ABCD$. This energy is exhausted when the area DEF equals $ABCD$, and the piston must then stop. If a longer stroke is desired, a later cut-off as at C' brings the piston to rest at F' (area $ABC'D'$ equals $D'E'F'$). For a quicker stroke, an increase of the working pressure with cut-off at C'' gives the same length of stroke as in the first case but at a higher rate. Admission and exhaust pressures are controlled by the regulating valves, Fig. 6.

SUMMARY OF ADVANTAGES

a A decrease in breakage of the sucker rods due to the decrease in maximum stress, to the decrease in the range of stress, and to the elimination of abrupt changes in the loading at the end of each stroke.

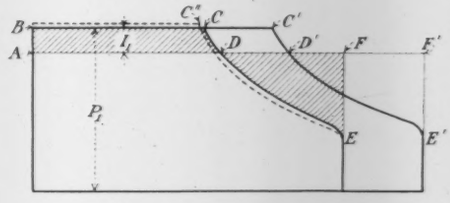

FIG. 12 ENGINE INDICATOR CARD

b A corresponding decrease in the stretching of the sucker rods during a cycle, proportional to the range in stress. This decrease is nearly 30 per cent, or three to four inches in length for a well 3000 ft. deep.

c The pump stops automatically if any unusual load is thrown upon it as when the well starts to "sand up."

d Friction in the actuating mechanism is greatly reduced. There is need for it. A series of tests recently carried out by one of our larger oil-producing companies showed that the efficiency of the pump-actuating mechanism from switchboard to the "polish rod" at the well averaged less than 25 per cent. The best result obtained was 48 per cent and the poorest was 8 per cent. This is so atrocious as almost to defy belief. An inspection of the walking-beam mechanism shows that these losses occur at the motor the two belt drives, the four shaft bearings, the wristpin bearing, and the three bearings of the walking beam. The walking-beam bearings do not contribute any great proportion of this loss, as the amount of rubbing of the engaging surfaces is very small. The engine drive does away with all but these walking-beam bearings and substitutes the friction of the piston and crosshead, which is necessarily small, since there is practically no thrust against cylinder walls or crosshead guides. Fig. 6.

e A cheaper source of power.

The ideal arrangement is to have a central compressor plant with large compressors driven by internal-combustion engines using either natural gas or oil. The compressed air would then supply a battery of wells of about the same depth. The load factor is practically 100 per cent, an ideal condition for Diesel drive. The pressure in the low-pressure receivers should be slightly less than the lowest exhaust pressure from any engine, while the pressure in the high-pressure receivers should be somewhat greater than the highest working pressure. If the compressor worked between limits of 25 lb. and 100 lb. gage, the temperature increase during compression would be about 140 deg. fahr. The mechanical efficiency of the compressors would be 85 per cent, or better, and the heat loss and loss of available energy during transmission and at the regulating valve would be low. In fact, the behavior of the air during its cycle might be compared to the action of a spring

which is alternately compressed and then allowed to expand and do work. Computation shows that the radiation loss during transmission would be about what would necessarily be wasted because of the difference in the characteristics of the compressor and the engine.

f Finally, the method provides a means by which conditions within the well itself may be studied by the use of the indicator card. This is a field for investigation about which little is positively known. Guesswork should have no more place in the oil industry than it has in a modern power plant.

Discussion

L. C. HAMPTON[7] and L. W. Voorhees[7] contributed a written discussion in which they said that there was, perhaps, no mechanical operation in the world which was not being constantly offered improvements, so the endeavor to improve was surely not proof of the inadequacy of existing mechanical methods. It could truthfully be stated that while the present method of pumping oil wells was by no means perfect, it was, to a certain degree, satisfactory and would not be discarded until something better had been conclusively demonstrated. Demonstrated improvements, some of which had recently been adopted as standard, and others which were still in the course of experimentation by the various oil companies, consisted of changes in counterbalances and driving mechanisms, the latter including reduction gears, chain drives, compensated tooth sprockets, and improved reverse gear clutches.

The author had entered the subject with considerable thought for the underlying fundamentals. This was shown by his careful and rational mathematical treatment, and his very logical assumptions and conclusions. However, the discussers felt that he had overestimated the trouble caused by breaking sucker rods. Data compiled from the experiences of two large oil-producing companies indicated that the major pumping troubles were, in order of importance:

1	Unscrewing of sucker rods	65 per cent
2	Breaking of sucker rods	18 per cent
3	Miscellaneous; such as tubing troubles, sanding up, etc	17 per cent

It would be noted that the table showed but 18 per cent of the pumping trouble was due to the breaking of rods and as this fact was based upon considerable data it could be accepted with good assurance. If the mechanism offered by the author should reduce the trouble caused by breakage by 75 per cent, it would reduce the total pumping trouble about 13½ per cent, which, of course, was desirable, but could scarcely be called revolutionary.

The author intimated that the stretch in the sucker rods of average wells was from 10 to 13 in. This was somewhat greater than the discussers' observations would indicate. They believed that a stretch of 2 in. per 1000 ft. was more nearly correct. In other words, 6 to 8 in. in a 3000-ft. well. If this was a true average of the stretch of the rods, his mechanism would only decrease the stretch about 2 or 2½ in., which would not be of any great advantage. For instance, the trouble experienced with gas entering the working barrel was greater than that which arose from the stretch of the rods.

The power transfer system in general use today was very inefficient, as was stated by the author. The discussers' observations, coinciding in general with his, were that the efficiency of this mechanism was about 21 per cent. By developing the actuating mechanism it was hoped to increase the power delivered to the polish rod to 40 per cent, in fact, recent experiences with a counterbalance on the bandwheel had raised the efficiency of the power transmission by some 10 or 15 per cent.

It must be remembered that there were other operations around a pumping well which required power. Power must be available at each rig for driving the calf wheel, sand reel, or bull wheel. Although the author had pointed out that a cheaper source of power was available for his air-operated mechanism, it was to be noted that the air compressor installation would be an additional charge to the equipment now on hand. Very roughly the discussers had calculated that a compressor of sufficient size to operate a well with 3000 ft. of sucker rods would cost about $1200 installed, assuming that a motor or gas engine was already at the well to drive the compressor. A large central compressor plant, including motor or steam engine, with distributing lines to ten wells could probably be installed at a cost of approximately $14,000 for motor, and $15,000 for steam, or about $1400 or $1500 per well. From their experience they have found that air was a very uneconomical medium for transmitting power in oil fields, as there was the usual loss in mechanical efficiency at the compressor and in the regulating valve, and in addition a considerable loss due to leakage of poor joints in pipe.

The author's mathematics showed interestingly and conclusively that the change of load at the end of each stroke was very abrupt and it was logical to assume with him that this abrupt change was responsible for much of the breakage of sucker rods. It was being found by a study of sucker-rod loads with dynamometers that the abruptness of the change could be greatly reduced with counterbalances. The dynamometers referred to were, in effect, registering balances which measured the load on the end of the beam throughout the stroke. It had been found that counterbalances were especially effective when mounted directly on the bandwheel and several oil companies were at present installing them in that position. One of the large oil companies using motors exclusively for pumping reported that a counterbalance on the pony beam had increased the efficiency about 20 per cent over that of wells with no counterbalances, and an additional 10 per cent increase in efficiency had been obtained by moving the counterbalance to the bandwheel.

In brief, the author had ably presented a modification to present methods of oil-well pumping offering several advantages, but which, in the minds of the discussers, scarcely justified itself.

In reply to Messrs. Hampton and Voorhees, the author said it was true, as they had said, that the endeavor to improve was not a proof of the inadequacy of a mechanism. No such thought had been intended, but a mechanism which the discussers admitted had an efficiency of but 21 per cent certainly furnished incentive to endeavor. Regarding the stretch of the rods during a cycle, he had studied many indicator cards taken from the polish rods of various wells. Some showed a variation between maximum and minimum stress of as much as five to one. An average was more than three to one. In many wells the maximum stress would seem to be close to the elastic limit. He was inclined to believe that his statement of the stretch was too conservative.

No one knew whether the plunger at the bottom of the rod had a greater or less stroke than at the polish rod. The two were not likely to be in exact phase. It was possible to conceive a condition such that the motions might be opposite at all times. Then all energy would be used to stretch the rod and would be dissipated to the oil. At the present time he was setting up a laboratory experiment which might throw some light on what occurs in the well.

The data which he had been able to collect, the author continued, would indicate that broken rods were a more frequent source of trouble than unscrewed ones when the rods had been properly set up. However, the cause of both troubles was the same, viz., vibration. This might be lessened in proportion to the decrease effected between maximum and minimum stress. Any motion that would decrease breakage would also lessen the unscrewing.

As he saw it, the essential thing was to permit the whole string of rods to float up and down like a free body so nearly as might be. If, when the heavy load of oil was abruptly thrown onto the rod there was at the other end a heavy actuating mechanism which moved without much regard to this loading, the stresses set up would be severe. If, on the other hand, the motion of the rod was determined solely by the way the load was applied, the stresses would be much more moderate.

What is said to be the largest piece of electrical machinery of its kind ever designed is a frequency changer which has been ordered by the Philadelphia Electric Co. from the Westinghouse Electric & Mfg. Co. This machine, when completed, will weigh more than 600 tons. It will occupy a space 55 feet long, and will be 22 feet high. At a 100 per cent power factor, the motor section of the unit will develop 60,000 horsepower. Twenty-five railroad cars will be required to carry the parts from the factory to the power plant.

[7] Engineer, Union Oil Company of California, Los Angeles, Cal.

The Selection of Fuel-Burning Equipment for Generating Stations[1]

By G. W. KEEN,[2] BALTIMORE, MD.

STOKERS and pulverized-fuel-burning equipment both have their advantages and disadvantages, and these must be balanced against each other when selecting a system for installation in a generating station. The two principal things to be considered when making such a choice are reliability and economy, and, in most instances, reliability is placed first. It is logical, however, to compare the two systems on the basis of economy to determine the more favorable one, and then decide whether or not the reliability features of that system are satisfactory. The more efficient system does not necessarily mean the more economical one—the primary consideration is the cost of producing steam.

Efficiency of Combustion

It is reasonable to expect that higher efficiency of combustion can be obtained with pulverized-fuel-burning equipment than with stokers for the following reasons:

1 Intimate mixing of the fuel and air makes it possible to operate with a smaller percentage of excess air, giving a higher percentage of CO_2 and a lower flue-gas temperature.

2 The air supply and fuel feed are easier to control, and the boilers are more adaptable to the use of automatic combustion control.

3 There is practically no loss due to unburned fuel in the ash.

4 Banking losses are much smaller, the only heat lost being that which is radiated from the setting.

5 Preheated air can be used to better advantage. Boiler efficiencies are being increased as much as 8 per cent by the use of preheated air, which can be applied both to stokers and to pulverized-fuel-burning equipment. It is reasonable to believe, however, that the use of preheated air under stokers will increase the number of stoker parts burned out. It is possible to prevent any considerable increase in furnace temperature due to the use of preheated air by the installation of water-cooled walls or radiant-heat superheaters, which can be designed to absorb an amount of heat from the furnace equivalent to that added by the preheated air. This condition, however, would not necessarily lower the temperature of the fuel bed itself, and there would be greater danger of burning the stoker parts. Some stokers keep the fuel bed broken up, thus reducing the liability of the formation of fixed clinkers under which destructive reverberatory action takes place, but they do not entirely eliminate the hazard. This condition, therefore, is a point in favor of pulverized-fuel-burning equipment which is not limited in any way by the use of preheated air, provided proper wall-cooling equipment is used.

Actual results now being obtained in central stations show that stokers, when using a good grade of coal, can compare favorably with pulverized-fuel-burning equipment at boiler ratings of about 150 per cent of normal rated capacity, but cannot show as good results at lower or higher ratings. When poorer grades of coal are used, stokers give lower boiler efficiencies at all ratings.

Cost of Installation and Operation

It is evident from the foregoing that, in so far as boiler efficiency is concerned, everything is in favor of pulverized-fuel-burning equipment. This advantage must be balanced against the extra cost of installing and operating such equipment, which involves:

1 Higher investment cost
2 More operating labor
3 More auxiliary power.

At first thought it would seem that the investment cost of the pulverized-fuel-burning equipment would be much higher than that of stokers, due to the cost of the coal-preparation plant. This extra cost is partly offset by the fact that the pulverized-fuel-fired plant requires fewer boilers for the same station capacity, due to the ability of the boilers to operate safely and efficiently at higher ratings. The maximum rating that can be carried on any boiler should not be limited by the ability of the forced-draft fans to supply air, or by that of the induced-draft fans to dispose of the gases. It is usually limited by the furnace temperature. It has already been pointed out that pulverized-fuel-fired boilers are more adaptable to carrying high furnace temperatures, and they can be depended upon to carry higher ratings than stoker-fired boilers, which allows a correspondingly lower rated capacity to be installed for the same steam requirements. This advantage, however, is not sufficient to balance the extra cost of the pulverized-fuel-burning equipment.

A pulverized-fuel-fired plant requires considerably more space than a stoker-fired plant. The use of unit pulverizers would partly offset this condition, but they are being used only on small boilers, and have not yet been developed to the stage where they can be economically employed on large boilers. If the land on which the plant is built is very expensive, or if the space available is limited, the stoker installation often proves the more economical.

The pulverized-fuel-fired plant requires less labor for operating the boiler room and for ash handling, but this is more than offset by the labor required for the coal-preparation plant.

When considering the extra cost of auxiliary power incident to the use of pulverized-fuel-burning equipment, it must be remembered that this extra auxiliary power reduces the station capacity by the same amount, so that the fixed charges on that amount of installed station capacity must be included.

There are no definite figures available at the present time to show which of the two systems should give the lower maintenance costs, but a study of these costs in a few large plants in this country, made by Hirshfeld, indicates a slight advantage in favor of pulverized-fuel-burning equipment.

Other Considerations in the Selection of Fuel-Burning Equipment

Size of Station. In large stations it is now good practice to install a small number of large boilers. The development of large stokers has not kept pace with the development of large pulverized-fuel-fired furnaces. No matter how good a stoker is, a certain amount of fire-tool work is necessary to keep the fuel bed in proper condition, and the larger the furnace, the more difficult becomes this task. Generally speaking, it can be said that pulverized-fuel-burning equipment has the advantage in large stations, with stokers holding the advantage in small stations.

Station Load. If the character of the station load is such that a plant must be operated at a low load factor, low boiler efficiency is obtained with stoker-fired boilers, due to the necessity of operating for a portion of the time at inefficient ratings, and due to high banking, stand-by, and pick-up losses. Stations which operate in conjunction with hydro stations, or which are used as peak-load or reserve stations, have this character of load. It would seem that pulverized-fuel-burning equipment would be better for such plants, but it must be remembered that these plants have a small output compared to the installed capacity, and the saving in fuel would be small compared to the increased fixed charges on the investment.

The Ash Question. A very important consideration is the ash question. In some localities it is objectionable to discharge a great amount of fine ash out through the stacks. This is sure to happen with pulverized-fuel-fired boilers unless precipitation equipment is used. It has been stated that from 40 to 70 per cent of the ash in the coal goes out through the stack, but the results of thirty continuous tests made at the Oneida Street Station in Milwaukee by the Bureau of Mines, under varying load conditions, show an average of about 29.5 per cent of the ash unaccounted for. If pre-

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[2] First Assistant Engineer, Consolidated Gas, Electric Light and Power Co., Baltimore, Md. Jun. A.S.M.E.

cipitation equipment is used, the problem of disposing of the ash presents itself. It cannot be handled like ordinary ash, due to its powdery form. When decisions are made on fuel-burning equipment for generating stations, the ash nuisance will often have sufficient weight to cause them to be made against pulverized-fuel-burning equipment, even though this system proves to be the more economical one to install.

Grade of Coal. One marked advantage which pulverized-fuel-burning equipment has over stokers is its ability to burn low grades of coal with good results. An attempt to burn low grades of coal on stokers results in poor combustion efficiency, increased boiler-room operating labor, and high stoker maintenance.

RELIABILITY

As regards reliability, stokers have one advantage over pulverized-fuel-burning equipment: the fuel bed contains a considerable amount of stored energy, which serves to help in maintaining the steam pressure in case of a sudden demand for steam, or failure of auxiliary power supply.

Pulverized-fuel-burning equipment has the marked advantage of containing no moving parts or grates in the furnace. It contains other mechanisms even more complicated than stokers, but they are located outside the boiler setting where they are accessible to the extent that they may be repaired while the boilers are still in service, and where other machines can be substituted for them while they are being repaired. It is true that a stoker installation possesses the greater simplicity because there are fewer "links in the chain," but this condition does not make it more reliable for the above reason. Considering the ability of the two systems to produce steam under all conditions at all times, it may be concluded that the pulverized-fuel-burning equipment is somewhat more reliable than stokers.

The Theory of Supersaturated Steam

BY DOUGLAS C. TURNBULL, JR.,[1] BALTIMORE, MD.

THE advocates of the theory of supersaturation as applied to steam, who are mainly British, believe that rapidly expanding steam, possessing an initial condition of dry saturation or superheat, does not remain in a state of thermal equilibrium throughout the whole range of expansion; but that at a certain interval during the expansion is in an unstable or supersaturated state. All the advocates of this theory believe that the steam expands as a perfect gas not only in the superheated region but also for a short period of time in the wet saturated region, in which region it is in the supersaturated form. Differences of opinion arise when British engineers try to decide what the condition of the steam is during the rest of the expansion. Most supporters of the theory believe that at the end of this metastable period (which has definitely been fixed by Wilson) the steam instantaneously becomes stable and completes the remainder of the expansion in thermal equilibrium. Others say that the metastable steam never does become stable again until the end of the expansion, and that its quality lies between that of supersaturated steam as it is at the Wilson point and that of steam in thermal equilibrium.

ORIGIN OF THE THEORY

In simple experiments in the flow of steam through nozzles, it was found that the observed flow was greater than the theoretical flow, based on isentropic expansion. In the case of a nozzle where friction and eddying may be expected to influence the discharge, it might have been anticipated that the actual discharge would be less than the theoretical. Yet in numerous experiments performed by different individuals the actual discharge was always from 2 to 4 per cent greater than the theoretical. The theory of supersaturated steam arose as an explanation of this discrepancy.

Considering similar pressure and entropy conditions for the comparison, it is found that the supersaturated steam has a smaller volume and a lower temperature (it is "undercooled" steam) than steam in thermal equilibrium. Its density and total heat are greater and it possesses zero moisture content, or nearly so. If the comparison were made with similar temperature conditions the supersaturated steam would be found to have a higher density and pressure and a smaller volume.

EXPERIMENTAL EVIDENCE IN SUPPORT OF THE THEORY

Having considered the theory of supersaturation very briefly, let us see what evidence supports it, and review the literature of the subject. In 1881 Aitken presented a paper on the results of a number of fog-formation experiments. He merely demonstrated the fact that condensation could not take place unless nuclei were present to start it. Lord Kelvin also explained the importance of just such nuclei, and stated that condensation could only take place on centers that had definite dimensions. Later Callendar considered that some of the co-aggregated molecules of the substance are of themselves capable of acting as nuclei for condensation, without the presence of nuclei proper. He estimated that the equivalent radius of the nuclei corresponding to the fine misty condition was of the order of 5.0×10^{-8} cm.

In November, 1897, Callendar and Nicolson performed similar experiments, using steam as the medium, and studied the condensation effect in the steam-engine cylinder. The observations recorded showed that the steam was distinctly supersaturated as it expanded behind the moving piston. In this same year Wilson experimented, using wet air as the medium, and concluded that there was a *definite* limit to the supersaturated condition, condensation taking place when the pressure of the fluid bore a certain ratio to the saturation pressure corresponding to the temperature. Hence there has been placed on the total-heat-entropy diagram what is known as the "Wilson line."

For the next ten years Stodola devoted most of his time to experimenting on the subject. In 1903 Lorenz, in going over some of Stodola's experimental results, applied the idea in connection with the expansion of steam through nozzles. Stodola continued his work, placing nozzles in his apparatus just as Lorenz did. The results of these tests emphatically convinced him and many others that the old idea must be abandoned and supersaturation considered. He came to this conclusion because of the excessive flow of the actual discharge over the theoretical, and because of the undercooling effect.

After 1903 ten years elapsed before anything further was published on the subject. In 1913 Professor Henderson presented a paper entitled Undercooled Steam, but no new facts were included in his report. Two years later Callendar presented a paper—On the Steady Flow of Steam through a Nozzle or Throttle—which included a history of his work from 1890 to 1915, covering the flow of steam in steam engines, simple nozzles, and steam turbines. It was an exhaustive study, and in addition to experimental results included the equations and sample calculations considering this metastable theory that apply to nozzle calculations and turbine design.

In 1918 Martin wrote an article entitled A New Theory of the Steam Turbine. In this he extended the application of the idea, applying it to actual turbine operations, and demonstrated that this supersaturated assumption rationalized the well-known "superheat corrections." It was not until this time that the Wilson line was introduced on the total-heat-entropy diagram. This line lies below the dry saturated line, and a definite region is included between these two lines which might be called the "Supersaturated Area." In this area the degrees of undercooling are numbered just as the degrees of superheat are numbered in the superheat region. Below the Wilson line on this total-heat-entropy chart equilibrium

[1] With Engineering Department, Consolidated Gas, Electric Light & Power Co., Baltimore, Md. Jun. A.S.M.E.

Presented at a meeting of the Baltimore Junior Local Section of the A.S.M.E., April 14, 1926.

conditions exist. The chart is therefore made up of three zones: namely, the superheated, the supersaturated, and the wet saturated.

The advocates of the theory disagree, as stated before, in regard to the properties of the expanding steam in the wet saturated region. Those who believe that the steam never becomes stable until the end of the expansion is reached are in a bad predicament, because they cannot state mathematically or otherwise what should be the properties of the steam below the Wilson line. Hence all the English textbooks include the Wilson line and consider the supersaturated steam to become stable at this line and expand in thermal equilibrium to the final pressure. In the last part of Martin's paper he states emphatically that it is his belief that steam expanding through a turbine never does regain equilibrium after the Wilson line has been passed. To prove this statement he cites several examples where the temperature at the turbine exhaust was below the temperature corresponding to the exhaust pressure. However, true readings of temperature and pressure at the exhaust are so difficult that this argument is not absolutely convincing.

In 1920 Callendar confirmed and added to his 1915 presentation in a treatise on The Properties of Steam. In the same year Ewing, in Thermodynamics for Engineers, in the chapter on jets and turbines, gave a very clear explanation of the rapid expansion of steam. Ewing, however, does not consider the steam to become stable at the Wilson line but at the end of the expansion. The change is considered to take place at constant pressure, accompanied by the giving up of some heat. The instant before the unstable steam becomes stable its pressure and volume will correspond, but its temperature will be lower and its total heat higher. During the next instant, when the steam actually becomes stable, there will be a loss of total heat and rise of temperature.

The most recent contributors to the subject are Mellanby and Kerr, who have printed four pamphlets presenting mostly test results of steam flowing through nozzles which they obtained during a three-year period ending in 1923, and including very little mathematics and theory. Turbine designers have criticized these four papers most severely because turbine nozzles were not used. Also in several sets of experiments, due to excessive outlet pressures, the ratio of throat pressure to initial pressure was not approximately one-half as it is in the case of turbine nozzles. The theory of supersaturation was confirmed by these experiments because of excessive measured steam flow.

Characteristic Coefficients for Hydraulic Machines

By BRUNO ECK,[1] AACHEN, GERMANY

TWO objections may be raised against the use of the characteristic (or specific) speed as a basis for comparing water turbines, centrifugal pumps, and other hydraulic machinery. These are:

1 The characteristic speed is *not* non-dimensional, which is desirable neither from the practical nor the theoretical point of view, as it is particularly troublesome to have to work with dimensional constants when comparing tests for which results are given in different systems of units, e.g., British and metric; and

2 It does not indicate the fundamental relationship that exists between turbines, centrifugal pumps, airscrews, screw propellers, windmills, etc.

The purpose of this note is to point out this fundamental relationship and to derive non-dimensional coefficients related to the characteristic speed from physical considerations.

Characteristic Speed of Hydraulic Turbines and Centrifugal Pumps

"Turbine runners of different types may possess characteristics of power, speed, dimensions, and operating head which vary widely. To make available a common basis for comparison the value designated as 'characteristic speed' is used. The characteristic speed of a runner (sometimes designated 'specific speed') is the speed in r.p.m. which a model of a runner would have if operated under a head of 1 ft., this model to be reduced proportionately in all dimensions from the original model until it will develop 1 hp. under 1 ft. head." [2]

Since the angles of the buckets of a runner are fixed, the linear speed of the bucket should at all times bear a constant relation to the linear velocity. The r.p.m. (n) of a runner is therefore proportional to $\sqrt{H}$, where H is the head in feet, and the speed at 1 ft. head ($= n/\sqrt{H} = n_1$) is called the unit speed. The power of a runner (N) is a function of the pressure and quantity of the operating water, and is therefore proportional to $H \times \sqrt{H}$, and the power of a runner at 1 ft. head is equal to $N/H\sqrt{H}$, and is called unit power (N_1).

The characteristic speed $n_s = n_1\sqrt{N_1}$

$$= \sqrt{N/H\sqrt{H}} \times n/\sqrt{H}$$

$$= n\sqrt{N}/H^{5/4}$$

For centrifugal pumps the formula is

$$n_s = 8.9n\sqrt{Q}/H^{3/4}$$

where Q is the weight of water discharged.

If n_s is held constant, the efficiency of the turbine or pump remains constant, or is a function only of the characteristic speed. In considering efficiency for geometrically similar machines, the Reynolds Number should theoretically be a constant. But this refinement in the application of the laws of similarity is scarcely of importance in practical engineering.

Similar Coefficients for All Hydrodynamic Machinery

It would appear that similar non-dimensional coefficients should apply to all forms of hydrodynamical machines, since the fundamental physical action is always the same: momentum is always received from solid bodies and delivered to liquids or gases, or received from liquids or gases and delivered to solid bodies.

In the theory of airscrews and of ship propellers two non-dimensional coefficients are employed:

1 Coefficient of thrust: $\sigma = \dfrac{\text{thrust}}{\text{dynamic head} \times \text{area}}$

2 Coefficient of pitch: $\lambda = \dfrac{\text{forward speed}}{\text{rotational speed}}$

We shall investigate whether analogous coefficients may be derived for turbines.

Coefficients for Axial-Flow Machines

For the hydraulic turbine, then, it follows by analogy that if

g = acceleration due to gravity in ft.
γ = specific weight in lb. per cu. ft.
H = head in feet
Q = quantity of water discharged in cu. ft. per sec.
n = r.p.m.
c_m = forward speed
u = peripheral velocity, and
D = diameter of runner,

then a coefficient of thrust may be written as:

$$\sigma = \frac{\gamma H}{\frac{\gamma}{g} \times \frac{1}{2} c_m^2} = \frac{2gH}{c_m^2}$$

[1] Aerodynamic Institute.
[2] Marks' Mechanical Engineers' Handbook, p. 1151.

The following relationships also obtain:

$$c_m = \frac{Q}{\frac{\pi}{4} D^2}; \quad D = \frac{u}{n} \frac{60}{\pi}; \quad \text{and} \quad \frac{u^2}{g} \nu = H$$

(The coefficient λ will be discussed later.)

Making appropriate substitutions,

$$\sigma = \frac{\pi^2 \times g^3 \times 60^4 \times H^3}{\nu^2 \times 8Q^2 \times n^4 \times \pi^4} = \frac{CH^3}{\nu^2 Q^2 n^4}$$

where C is a constant.

Taking the characteristic-speed equation of the centrifugal pump

$$n_s = 8.9 \frac{\sqrt{Q}}{H^{3/4}}$$

and substituting, we find

$$\sigma\nu^2 = \frac{240 \times 10^{11}}{n_s^4}$$

The characteristic speed is thus seen to be inversely proportional to the fourth root of $\sigma\nu^2$.

Coefficients for Radial Runners

In similar fashion it is possible to introduce a coefficient of thrust for radial runners. In Fig. 1 we have a radial wheel and an axial runner of the same outside diameter. If the two discharge the same weight with the same head,

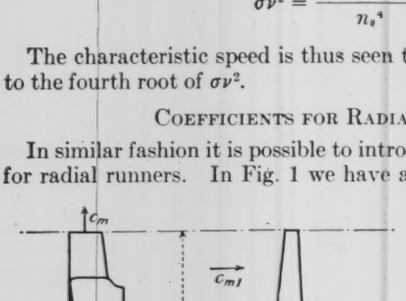

FIG. 1

$$c_m \pi bD = c_{m1} \frac{\pi}{4} D^2$$

and

$$c_{m1} = 4 \frac{b}{D} c_m$$

These two runners have the same coefficient of thrust, and we have only to make appropriate substitutions for the radial runner to obtain

$$\sigma = \frac{\gamma H}{\frac{\gamma}{2g} {c_{m1}}^2} = \frac{\gamma H}{\frac{\gamma}{2g} 16 \left(\frac{b}{D}\right)^2 {c_m}^2} = \frac{D^2 H}{8 b^2 {c_m}^2}$$

Writing

$$c_m b \pi D = Q; \quad D = \frac{u}{n} \frac{60}{\pi}; \quad \text{and} \quad \frac{u^2}{g} \nu = H,$$

gives

$$\sigma = \frac{\pi^2 g^3 60^4 H^3}{\gamma^2 8Q^2 n^4 \pi^4} = \frac{240 \times 10^{11}}{\gamma^2 n_s^4}$$

That is, the coefficient of thrust is of identical form and the characteristic speed is again inversely proportional to the fourth root of $\sigma\nu^2$.

Three Possible Non-Dimensional Coefficients

It is possible to write down three non-dimensional coefficients:

$$C/c_{m1}; \quad C/u; \quad \text{and} \quad c_{m1}/u$$

If $C = \sqrt{2gH}$, $c_{m1} = \frac{Q}{\frac{\pi}{4} D^2}$, and $u = \frac{\pi n D}{60}$, these coefficients become

$$\left.\begin{aligned} \sigma = \left(\frac{C}{c_{m1}}\right)^2 = \frac{2gH}{{c_{m1}}^2} \\ \nu = \frac{Hg}{u^2} \end{aligned}\right\} \text{Coefficients of thrust}$$

$$\lambda = \frac{c_{m1}}{u} \qquad \text{Coefficient of pitch}$$

Since $\lambda = \sqrt{2 \frac{\nu}{\sigma}}$, only two of these three coefficients can be written down independently. If we select σ and ν, we get (since $c_{m1} = Q/(\pi D^2/4)$ and $u = \pi n D/60$):

$$\sigma = \frac{g\pi^2 H D^4}{8Q^2}$$

and

$$\nu = \frac{g}{\pi^2} \frac{60^2 H}{n^2 D}$$

A non-dimensional coefficient which shall fully determine the characteristics of a turbine should not contain the term D. Such a coefficient is $\sigma\nu^2$, since

$$\sigma\nu = \frac{\pi^2 g^3 60^4 H^3}{8Q^2 n^4 \pi^4} = \frac{240 \times 10^{11}}{n_s^4}$$

It may be convenient to use an expression

$$\psi = \sqrt[4]{\frac{1}{\sigma\nu^2}} = \frac{n_s}{2210}$$

The advantages of using $\sigma\nu^2$ are, first, that it is non-dimensional, and secondly, that it relates all types of hydrodynamic machinery. This should be particularly valuable in the case of the Kaplan propeller turbine.

Whether it is expedient to introduce this non-dimensional coefficient is ultimately a matter for practical engineers to decide.

Arc Welding for Building Construction

MARKING a distinct step in the engineering study of the use of arc welding for building construction, the Westinghouse Electric & Mfg. Co. on July 28 and 29 staged a demonstration in the materials laboratory, Machinery Hall, Carnegie Institute of Technology, Pittsburgh, which is bound to stimulate the interest of steel fabricators, contractors and engineers in the subject, because up to this time nothing quite so comprehensive or informative as this series of tests had been attempted. There was a registration of about 100 men, representing almost that many companies engaged either in the actual fabrication of structural steel or in its erection or design. No fewer than 22 tests were made on a vertical testing machine with a capacity of 400,000 lb.

Engineers of the Westinghouse company for some time have been studying the uses of arc welding and very early in their studies arrived at the conclusion that structural steel offered the broadest field for development. It was not long afterward that it was found that in order to gain all of the advantages of the process, beams, columns, and joints should be designed especially for welding and that welded sections based on standard riveted practice had little if any merit over riveted members. It was frankly stated by Dr. G. D. Fish, structural engineer, New York, who has been retained by the Westinghouse company as a consulting engineer on arc welding for building construction, that research had not yet gone far enough to disclose the minimum of welding that could be done and insure the same strength as riveted steel and that in the beams, columns and joints for the tests there had been probably a stronger and heavier weld than later studies would disclose had been necessary.

It was brought out during the tests that the welding steel was of ordinary mild steel analysis; that the electrodes were not coated; and that the welding had not been done by specially trained welders. The American Bridge Co. coöperated with the Westinghouse engineers in the studies.

Sustaining its engineers in their findings was an announcement that the Westinghouse company was to build two arc-welded structural-steel buildings. One of them is to be a one-story building to be used as an engineering laboratory at East Pittsburgh, and the other a five-story mill-type building at Sharon, Pa., to be part of the transformer plant. A feature of the East Pittsburgh building is that it will be erected partly of scrap roof trusses, this in emulation of a plan followed in the construction of a building for the Chicago, Burlington & Quincy Railroad at Eola, Ill. (The *Iron Age*, vol. 118, no. 6, Aug. 5, 1926, p. 344.)

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

The Science of Grinding

AS THE science of grinding is one which is more or less directly connected with chemistry, physics, mechanics, and mathematics, the literature is so scattered and so obscure that it is very difficult to find out what has been achieved in the subject. The work detailed below was carried out between 1922 and July, 1925, and establishes the right of grinding to rank as an exact mechanical science, with her own laws and methods of calculation, as definite as those appertaining to electricity or steam. In other words, just as the engineer knows the amount of electrical energy or steam necessary to perform a given amount of work under definite conditions, a development of this research would enable one in the future to estimate the theoretical amount of work required to reduce a given material to a given degree of fineness under given conditions. It will thus be possible to deduce the efficiency of any grinding machine.

The importance of this development in the gold mining, pigment, cement, ceramic, and other industries—which depend so largely upon the economical production of materials in a state of fine division—will be apparent when one reflects on the great advances which occurred in the sciences of electrical and steam engineering after the underlying laws were discovered.

The work was carried out under most worrying and discouraging circumstances. The Government withdraw their grant long before the British Portland Cement Research Association was on its feet, while the cement trade itself was openly hostile. This was all the more remarkable because the cost of grinding the 4,000,000 tons of cement annually produced in the United Kingdom must run to well over £1,000,000 annually. When we consider the enormous sums of money spent on grinding not only by the cement industry but also by other industries as well, such as mining, metallurgical, coal, pigment, fertilizer, cereal, explosives, and others too numerous to mention, to all of which the subject was of vital importance, it is curious that no effort was made by any of the interested trades to help or encourage the work.

The following is a brief summary of a few of the results obtained, using pure quartz sand as the easiest substance to work with, since it is a merely chemically pure, brittle, crystalline substance, which can be chiefly obtained in unlimited amounts and whose surface can be accurately measured by chemical laws.

LAW 1 (Rittinger's Law). *The surface produced is accurately proportional to the work done. Double the work and the surface produced is doubled. Treble the work and the surface produced is trebled.*

As a result of these experiments it is a simple matter to calculate the work required in grinding to any requisite degree of fineness with any given grinding media.

$$W = B\,(S_2 - S_1)\ \text{ft-lb} \qquad [1]$$

where W = the work required in ft-lb.; S_2 = final surface of powder in sq. ft.; S_1 = original surface of powder in sq. ft.; B = a constant peculiar to the grinding medium employed, the material ground, and the type of grinding machine employed, being the work required to increase the surface by 1 sq. ft.

Using 1-in. steel balls in the 18-in. mill for quartz sand, B was found to be 60.9 ft-lb.

Example. Calculate the power required to grind 3 tons of quartz sand per hour from a fineness represented by 24.3 sq. ft. per lb. (Leighton Buzzard standard cement sand $-20^2 + 30^2$ cement sieve) to a fineness represented by 827 sq. ft. per lb. (corresponding to 1 per cent on the 200 I.M.M. sieve), using 1-in. steel balls in an 18-in. × 18-in. mill.

3 tons = 6720 lb.

Final surface $S_2 = 6720 \times 827 = 5{,}558{,}000$ sq. ft.

Initial surface $S_1 = 6720 \times 24.3 = 163{,}000$ sq. ft.

Constant $B = 60.9$ ft-lb. Hence, employing formula [1],

$$W = 60.9\ (5{,}558{,}000 - 163{,}300)$$
$$= 328{,}600{,}000 \text{ ft-lb.}$$

1 hp. = 33,000 ft-lb. of work per minute
= 1,980,000 ft-lb. of work per hour

Hence theoretical horsepower required for grinding 3 tons per hour is $328{,}600{,}000 \div 1{,}980{,}000 = 166$ hp.

Further work indicated that there is a thermodynamical connection between the work required in grinding and the total heat required to volatilize the substance. This is fairly possible, for, considering the simplest possible case, in gasifying a homogeneous solid material the substance is reduced to particles of molecular dimensions, whereas in ordinary grinding the same action is performed

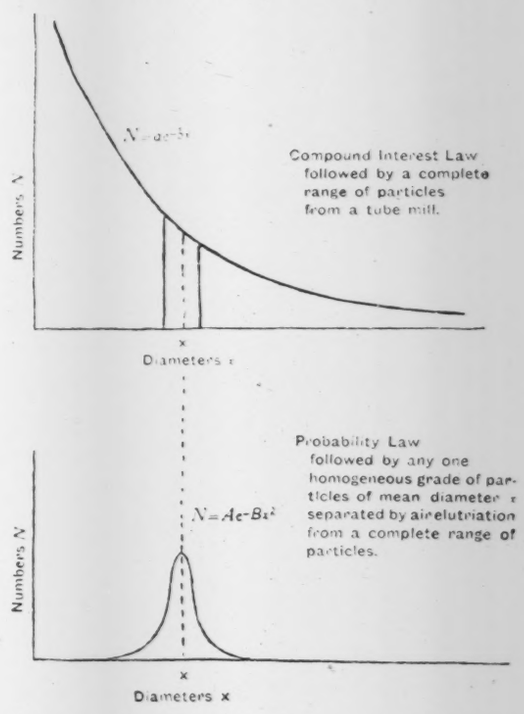
Compound Interest Law followed by a complete range of particles from a tube mill.
Numbers N
Diameters x
Probability Law followed by any one homogeneous grade of particles of mean diameter x separated by air elutriation from a complete range of particles.
$N = Ae^{-Bx^2}$
Numbers N
Diameters x

FIG. 1 CURVES OF COMPOUND-INTEREST LAW AND PROBABILITY LAW AS APPLYING TO GRINDING

but the particles remain of considerable size. Because of this the absolute efficiency of a grinding machine can be deduced by grinding brittle crystals of known heat of volatilization.

The following calculation is interesting. Imagine quartz sand to be continually ground until reduced to particles the size of molecules. Then the work required to produce this result is the same as that required to convert the silica into a gas (where each molecule is separated from the other). So that the work of grinding will be the same as the work required to volatilize the substance. From Bragg and Milligan's recent work we may calculate the surface exposed by 1 lb. of silica molecules as 38×10^6 sq. ft. From Troutman's rule we may calculate the latent heat of evaporation of silica as 1765 B.t.u. per lb., so that the work required to gasify 1 lb. of silica is $1765 \times 778 = 1.4 \times 10^6$ ft-lb. Therefore to increase the surface by 1 sq. ft. requires $(1.4 \times 10^6) \div (38 \times 10^6) = {}^1/_{27}$ ft-lb. The value required in practice with 1-in. steel balls is 60 ft-lb. Hence the efficiency of a modern tube mill is only ${}^1/_{27} \times 100 \div 60 = {}^1/_{16}$ per cent. We thus have still a long way to go in making grinding machines efficient.

The grinding efficiency should be measured in terms of the work in foot-pounds required to increase the surface of a standard quartz sand by 1 sq. ft.

LAW 2. *The number of particles produced increases with decreasing diameter, according to the compound-interest law.*

In other words, the number of particles grows with increasing fineness in the same way that money grows when let out at compound interest.

Mathematically the law may be stated thus: In a given weight of finely crushed sand, if N be the number of particles of diameter x, and if N and x be considered as variables, then in every case so far tested,

$$N = a\,e^{-bx} \quad [2]$$

where a and b are two constants characteristic of the particular sample treated (Fig. 1).

Differentiating [2] we obtain

$$\frac{dN}{dx} = -b\,a\,e^{b\,-x} = -bN \quad [3]$$

In other words, the rate of increase with decrease of diameter in the number of particles of a given degree of fineness is proportional to the number already present of that particular degree of fineness.

If we take logarithms of [2] we get

$$\log_e N = \log_e a - b\,x$$

so that if we plot the logarithms of the numbers against diameters we obtain a straight line.

In the original article a table is given representing some typical results and it is shown that when the logarithms of the numbers are plotted against the mean diameters of particles, a straight line is obtained.

LAW 3. *The average shape of the particles produced in crushing remains the same whether they are large or small.*

This was proved by grading out the crushed quartz sand by elaborate air-elutriation experiments, and then proceeding to measure the absolute diameter, surface, weight, and number of the particles in each grade. The result was confirmed by plotting against the diameters the terminal speeds obtained when particles of sand of different diameters were allowed to fall in air at a constant temperature. Both the small and the large particles occurred on the same straight line, which would not have been the case if their shapes had differed.

If S be the statistical surface of the particles of statistical diameter d, then S/d^2 = constant A, no matter whether the particles are of large or small size. This constant A it is proposed to call the "surface constant."

A table in the original article represents the results obtained from crushed sand. The variations from constancy are within the limits of experimental error when the extreme difficulties of the determination are considered. It is shown that for sand particles ground in a tube mill, and of statistical diameters ranging from 0.00333 cm. to 0.01089 cm., this surface constant is about 2.

In other words, if the statistical diameter of a particle be d cm., the statistical surface of this particle will be $2d^2$ cm.

LAW 4. *Any grade of crushed sand coming from a tube mill is composed of homogeneous grades of crushed sand in which the distribution of the numbers of the particles with their diameters cannot be altered, no matter how often we regrade the sand.*

These statistically homogeneous grades of sand may be considered as the unalterable elements which compose a given mixture of sand particles. In such a homogeneous grade the distribution of the particles follows the probability law.

This was proved by separating out the sand coming from a tube mill (and whose particles followed the compound-interest law $N = ae^{-bx}$) into separate grades by repeated air elutriation. One of these grades was then elutriated some eight or nine times without in any way altering the distribution of particles therein, which was found to follow the probability law: $N = ae^k$, where N is the number of particles of a given diameter x, $k = -Bx^2$, and a and B are constants.

This law is also probably a mathematical consequence of Rittenger's law and the compound-interest law. The laws regulating particle size are established by this research. (Geoffrey Martin, Late Director of Research of the British Portland Cement Research Association, in *Chemistry and Industry*, vol. 45, no. 23, June 4, 1926, pp. 160 T to 163 T, 3 figs., *et.A*)

Short Abstracts of the Month

CORROSION

Corrosion Symposium of the Institute of Chemical Engineers

AMONG the papers presented at the meeting of the American Institute of Chemical Engineers, June 21–24, 1926, was one by F. F. Chapman of the Du Pont Co. dealing with the ability of chromium steels of the stainless-steel type to resist corrosion by nitric acid at all temperatures up to the boiling point. The author claims that these steels are particularly resistant and that welded joints in chromium-steel equipment are satisfactory if the correct technique of welding has been used. Apparently the use of chromium steels of different compositions in the same line is permissible, although occasional failures have been obtained under these conditions.

Another paper devoted to substantially the same subject was one by C. E. MacQuigg of the Union Carbide and Carbon Research Laboratories. The author said that chromium imparts oxidation resistance to the various alloys, and this holds true for oxidation in general, either wet or dry and at low or high temperatures. The resistance of the element chromium and its alloys to chemical attack by electrolytes resides in the inertness of the product formed as a protective film. Nitric acid is without effect if the metal contains about 12 per cent or more of chromium. For resistance to wet attack in other than oxidizing media no general rules can be formulated. Against sulphuric acid the ordinary chromium-iron alloys are not reliable under all conditions, although some of the installations have been successful. They are easily attacked by hydrochloric acid. Sulphur and sulphur gases are practically without effect on straight high-chromium alloys. The presence of nickel, however, is to be avoided due to reactions with sulphur.

Paul D. Merica of the International Nickel Co. outlined the progress now being made in corrosion research. Present tendencies are against the so-called universal alloys capable of satisfying widely varying conditions of service. Instead alloy types are being developed for specific purposes. This plan is said to have yielded satisfactory results. (Abstracted through *Chemical and Metallurgical Engineering*, vol. 33, no. 7, July, 1926, pp. 397–401, *g*)

ENGINEERING MATERIALS

The Mechanism of Spalling

STUDY has been made of the stresses and fractures developed in a solid when rapidly heated or cooled. The stresses were measured in bakelite specimens by the photoelastic method. High stresses in all cases were found to occur near the surface of the solids, but no high tension stresses were present when the specimens were being heated. A number of vitreous-clay spheres and bricks were heated and cooled rapidly and the nature of the cracks was noted. In practically every case the fractures occurred as predicted from the measured stresses. Torque-deflection curves were obtained for a number of firebrick in torsion at different temperatures. The temperature at which plastic flow occurs for every material tested was approximately the same as the temperature of the initial deflection under compressive loads in a standard load test.

From preceding investigations the following conclusions are derived:

1 Spalling fractures are caused both by shear and tension stresses.

2 Spalling which occurs on rapid heating is due entirely to shear stresses.

3 Spalling on cooling is due mainly to tension stresses and, to a small extent, to shear stresses.

4 There are no large tension stresses on sudden heating.

5 The shear stresses are about equal on either heating or cooling, and are about one-half the value of the maximum tension or compression stresses.

6 Shear fractures occur suddenly near the edges and corners of the solid, and cracks enter the original surface on the lines of shear, that is, at about 45 deg. to the surface. Close to the edges and the corners the fractured surface is curved slightly like the lines of shear stress.

7 Tension fractures appear across the bricks in parts remote from the corners due to concentration of tension stresses at these points. They start at the surface and penetrate deeper and deeper into the solid in each cycle of cooling. In general, the fractures follow the lines of principal stress and enter normal to the surface.

8 Fireclay bricks show plastic flow in shear at approximately the same temperature as they show deflection in a compression load test. (F. H. Norton, Babcock & Wilcox Fellow, Division of Industrial Coöperation and Research, Massachuetts Institute of Technology. *Journal of the American Ceramic Society*, vol. 9, no. 7, July, 1926, pp. 447–461, 20 figs., *te*)

FUELS AND FIRING

Report of the Canadian Peat Committee

THIS is a report of a committee appointed jointly by the governments of the Dominion of Canada and the Province of Ontario, and published by the Mines Branch, Ottawa.

The Committee came to the conclusion that the air-dried machine peat process is the only one in successful commercial operation, or even giving the slightest promise of success. The Committee therefore eliminated all methods or processes which depend on artificial means for the separation of moisture and concentrated their effort on the devising of ways and means for carrying out in the most economical manner possible the various stages of the air-dried machine peat process.

The Committee tried out two types of machines, the Anrep and the one devised by E. V. Moore. It was found that the Anrep excavating element was in actual operation superior to the Moore plant, but the Moore spreading system proved to be far more efficient. As a result a combined plant was built, and tried out during the season of 1922.

Since the establishment of a peat-fuel industry will naturally raise the question of disposal of those areas of the peat bogs not worked for fuel production as well as those from which the fuel peat is in the course of time removed, a chapter on agricultural uses of peat and peat lands has been included in the report together with reports on special investigations.

The general conclusion of the report appears to be that a peat-fuel industry can be established in Canada on a sound economic basis.

The investigation of the various problems concerned has not yet been completed and the following subjects are still under consideration, namely, the use of peat as a substitute for hardwood in the wood-distillation plants, and the briquetting of peat with and without the addition of coals and with various types of binders. (*Canadian Mining Journal*, vol. 47, no. 29, July 16, 1926, pp. 707–709, *g*)

The Schuckert-Petri Method of Combined Powdered Fuel and Grate Firing

IN THE method here described part of the fuel is burned on the grate and part, the latter in the form of coarse powder or dust, is fired through a nozzle installed in the ignition arch above the traveling grate in such a manner that the dust flame plays on the layer of fuel near where the latter enters the combustion chamber. By means of the dust flame the fuel on the grate is ignited more effectively and dependably than by means of the ignition arch. Since the dust flame is directed on to the layer of coal, its destructive influence on the furnace walls is decreased. The dust flame unites with the grate flame above the grate and burns to completion since it is much longer than the grate flame.

The dust flame is so directed that the pieces of slag and the larger coal particles which it contains are thrown out of the flame and on to the grate, on which they are carried off or burned with the grate coal.

Apparently it is not the intention primarily to pulverize the fuel but merely to use fuel of very small sizes, part of which is already broken up to the required state. Moreover, the powdered fuel for the dust flame in this system may be much coarser than in exclusive powdered-fuel firing. The equipment for the boiler includes a "wind sifter," which is a simple pipe into which the coal containing grains up to roughly a quarter of an inch is poured at the top. A current of air flowing upward strikes this coal and carries the finer particles into the dust flame, while the coarser ones drop from the lower end of the pipe into the coal pocket ahead of the scale.

The author points out that in Germany various mines furnish fine coal leaving from 50 to 70 per cent residue on a 178-mesh-per-inch sieve. This coal is too coarse for ordinary powdered-fuel firing but can be used directly for supplementary powdered-fuel firing without further preparation, not even the use of a "wind sifter" being necessary.

The original article gives test results on boilers operating with and without supplementary coal-dust firing.

To date there is a total of 21 boilers in 11 different plants with a total heating surface of close to 85,000 sq. ft. equipped with this method of firing, and 14 more such furnaces are being built with a total heating surface in excess of 100,000 sq. ft. Some of these boilers have been in operation for three or four years. There was trouble at first with the side walls and bridgewalls, but this has been largely eliminated.

The main economy due to the use of the new furnace appears to lie in the fact that it permits using very cheap grades of coal for which there is no great demand at present. It would appear, however, that if a large number of big users should turn to these kinds of coals, the latter would increase in price, thus defeating the aim of the new type of furnace. (*Archiv für Wärmewirtschaft*, vol. 7, no. 2, February, 1926, pp. 39–44; translation in *National Engineer*, vol. 30, no. 7, July 1926, pp. 291–295, 4 figs., *d*)

Oil from Coal

ACCORDING to the author, almost impenetrable secrecy surrounds the processes for producing oils from coal. There are a vast number of workers in this particular field, but only now and then is a small corner of the veil lifted and some few facts and figures made available. Even these appear in a form which will not allow any definite conclusions to be drawn one way or another.

The first thing which confronts the investigator is the lack of system in the fundamental examination of oil, or, in other words, discrimination between oils and tars. Professor Fischer has indicated methods for this purpose, however. The oil products obtained by the straight distillation or carbonizing of coal can be divided up into three main groups: (*a*) Purely primary products; (*b*) a mixture of primary and secondary products; (*c*) purely secondary products.

Taking these groups one by one, we find that the primary products may be produced—

(*a*) By subjecting the coal to a hot current of gas—i.e., distilling by means of an inert gaseous heating medium of such volume that all the heat necessary for distillation is carried as sensible heat in the gas. External heating may also be employed, provided a large enough volume of gas or steam is continually sweeping through the retort. As examples of the two systems may be mentioned the "L & N" retort and the modified Stinnes retort.

(*b*) A mixture of primary and secondary products is obtained when any externally heated retort system is used, where no sweeping gas is employed, or when internal heat is used and oil products are condensing on the raw coal, being continuously reëvaporated as the coal proceeds onward. For example, the externally heated Thyssen retort and the internally heated Lurgi and Maclaurin retorts, or vertical gas-works retorts, run under steaming conditions.

(*c*) Secondary products are the order of the day in present coke-oven and horizontal gas-works practice.

There is a fourth way of obtaining oil products from coal, namely, that of extraction by means of suitable solvents under either heat or pressure, or both combined.

Investigators agree upon the vital importance of obtaining the so-called primary products—that is to say, oil products recovered with little or no secondary decomposition. Once the right and sure primary foundation is there, the erection of the subsequent

structure is much simplified. It was not until recently that a method of identification was worked out by Fischer and his collaborators referred to above. The method is based upon the observation that original primary products which have been subjected to thermal decomposition are, on subsequent distillation, far more easily decomposed, and alter in their behavior toward petrol-ether; in other words, once soluble in petrol-ether, they become on subsequent distillation only soluble to a degree. Besides this, a marked increase in specific gravity is noted.

The original article gives tables of properties of products of decomposition illustrating the remarkable difference between the primary oil and the vertical tar. Another highly interesting fact which is shown by the tables is that provided they are primary products in the correct sense of the word and have been obtained under the proper conditions and strict temperature control, all the neutral oils from bituminous coals are identical, irrespective of district, coking or non-coking, younger or older formation.

Consequently one may say with confidence that the true primary oil products obtained from very varying grades of bituminous coals are identical not only in composition but also in specific gravity, relative percentage by weight of fractions, and physical properties. The importance of these observations cannot be overrated, because they make it clear that by suitable treatment a uniform crude oil can be obtained from the various bituminous coals in the United Kingdom; and this opens up the possibility of producing standard oils throughout, the oils being amenable to the same refining treatment and miscible in all proportions. A series of investigations on primary oils from brown coals and lignites is at present being conducted by the author in order to correlate primary oils from these with those from bituminous coals.

Oil-distillation curves give some clues to the nature of the oils but not enough, and examination of the relative specific gravities and ultimate analyses becomes necessary, as the author shows by curves and tables.

He explains next why the vertical retorting system with internal heat application does not produce a true primary product, and asks what are the distillation conditions under which a true primary oil may be obtained. What follows is of such interest that it is reproduced verbatim.

Experience lays down the following rules:

1 Internal heat solely, or, if external heat is used, so large a volume of protecting gas or steam must be admitted that the oils set free do not get into contact with any surfaces or zones of temperatures higher than those at which the oils were evolved in the first instance.

2 The heating must be counter-current to the flow or passage of the coal.

3 The distillation products must be removed in strict uniflow with the heating-gas current.

4 Sufficient volume of gaseous heating or protecting medium must be present to lower the vapor tension of the heaviest fractions to such an extent that no condensation takes place within the retort at the temperature prevailing in the coldest zone.

5 Sufficient so-called free space or gas space must be available to secure a free passage of the oil vapors without these being forced to penetrate through colder layers of raw material.

6 Preferential preheating of the raw coal entering to temperatures above the dewpoint of the oil vapors.

7 The process of distillation must be continuous.

8 The process must be under complete temperature control.

9 The process should permit modifications in regulation of coal feed, duration of distillation, and volume of distilling medium, independent of one another.

10 The process should be such that the composition of the distilling medium, and the temperature, can be modified and controlled independent of the retorting.

11 The source of heat must be altogether separate from the retort.

12 The largest possible single units should be employed, in order to secure steady running conditions and uniformity of output.

An oil which has been produced under these conditions is the "L & N" oil, and, as remarked above, it is a standard product, varying very little indeed from coal to coal. This has been conclusively proved by the distillation on a large scale of many hundreds of tons of bituminous coal of the most widely varying descriptions.

A very remarkable property of the high-boiling "L & N" neutral oil is its great thermal stability. For instance, the fraction 350 deg. to 400 deg. cent. was maintained in an open Engler flask at 345 deg. to 348 deg. cent. for four consecutive hours, only showing an average loss in weight of 1.3 per cent due to evaporation, and there was no sign of thermal decomposition. The above test is fairly conclusive, as far as thermal stability is concerned, that neutral oils obtained by carefully conducted external-heat retorting show a profound change in ether solubility on being momentarily heated to 250 to 300 deg. cent.

From the "L & N" high-boiling oils, therefore, can be manufactured a lubricating base which will meet all the demands of, and tests applied to, the usual well-oil lubricants. Figures regarding tests carried out at the National Physical Laboratory, Teddington, in comparison with a highly priced, well-known oil for ascertaining the coefficient of static friction are given in the following tabulation:

	—Coefficient of Static Friction—	
Lb. per sq. in.	Comparative oil	"L & N" oil
20	0.129	0.114
40	0.129	0.115
60	0.126	0.116
80	0.129	0.118
100	0.128	0.117
120	0.131	0.116

The value of the coefficient obtained, under the same conditions, for rape oil was 0.080.

The author tells next of an investigation undertaken in Germany by Meyron Heyn and Manfred Dunkel in reference to the Bergius process. Because of the attention which the Bergius process received in this country the following conclusions drawn from Heyn and Dunkel's investigation become of interest.

1 The oil yield on the coal substance (ash- and moisture-free coal) is less than that claimed by Bergius, and aggregates 31.2 per cent by weight. The oil is calculated as pitch-free unrefined oil, and includes the phenols.

2 The crude oil contains a high percentage (34.8 per cent by weight) of dust, which is extremely difficult to separate out on a small scale, and will cause serious troubles on a working scale.

3 The phenol percentage is high, almost as high as that of vertical retort tar (12 to 22 per cent).

4 The ultimate analysis shows that the oils are essentially cyclic and aromatic compounds of benzol structure.

5 The refining losses are heavy (2.5 to 5 times that of the true primary oil ("L & N" oils).

6 The percentage of lubricating oil is negligible.

7 In spite of every endeavor, the investigations did not succeed in producing a lubricating oil which could stand up to any of the tests demanded of a good-grade oil.

8 The higher-boiling fractions, in spite of extensive refining, are not stable. Polymerization continues to take place, with the formation of gummy substances.

9 The motor-spirit fraction does not discolor on exposure to light, is distinctly good, and represents 20 to 25 per cent of the oils (including the light spirit scrubbed from the gas).

10 The inference can be drawn that the rest of the neutral oils can only be used as Diesel and fuel oils.

11 The phenols are apparently not hydrogenated to any extent, if at all, in spite of Bergius' former claims.

12 The phenols are principally of the lower series, or, in other words, resemble those found in high-temperature tar more closely than those generally prevailing in low-temperature oils. This is also evidence as to the secondary nature of the Bergius oils.

13 The paraffin percentage is very small.

14 It is clear from the above that from the crude Bergius oil it is not possible to obtain all the various grades required either for national or industrial purposes, as some of the most vital oils are missing. (Harald Nielsen in the *Gas Journal*, vol. 174, nos. 3291, 3292, June 16, 23, 1926, pp. 591–592, 650–653. The same article in *Iron and Coal Trades Review*, vol. 112, nos. 3041, June 11, 1926, pp. 919–920; in neither publication is the article complete. *etA*)

INTERNAL-COMBUSTION ENGINEERING

Combustion in Gas Engines

ON JUNE 14 a general discussion on explosive reactions in gaseous media was held under the auspices of the Faraday Society at the Institution of Mechanical Engineers in London. The program was divided into two parts, in the first of which explosive reactions were considered generally, and in the second, in reference to internal-combustion engines. Among others the two papers abstracted below were presented.

COMBUSTION IN GAS ENGINES

This paper, by Prof. W. T. David, discussed the extent to which incomplete combustion of the charge was responsible for limiting the pressures developed in gas engines, and analyzed the factors influencing the rate of combustion. From this survey of experimental work, the author said that there appeared to be little doubt that incomplete combustion ranked equally with increasing specific heat as a primary cause limiting the pressures developed in gas engines. Fortunately, after-burning did not affect the thermal efficiency of a gas engine to any serious extent provided that the ignition timing was arranged so as to give a reasonable vertical explosion line in the indicator diagram, for the bulk of the after-burning took place in the early stages of the expansion stroke; and, indeed, from the point of view of mechanical design it was an advantage in that it tended to reduce the ratio of the maximum pressure to the mean effective pressure.

Various factors influencing the rate of combustion in gas engines were discussed. Of them, the author referred particularly to turbulence, and said that the greater the degree of turbulence or eddying of the gaseous charge, the more rapidly would inflammation spread. Experiment showed that the time of explosion of a stagnant gas-engine charge in a closed vessel was very much greater than that of a charge of similar combination in the cylinder of a gas engine of which the combustion space had the same volume as the closed vessel. Indeed, were that not so, gas engines could not run at present-day speeds—at any rate, with the normal ignition advance. The reason why turbulence had such a great influence on the rate of burning would appear to be that, owing to the actual motion of the gaseous charge, parts of the portion first inflamed were carried to various quarters of the charge, and, in effect, a large number of ignition centers were set up. Experiments by Hopkinson showed that a degree of turbulence beyond a certain point had little influence on the rate of explosion. The experiments were made in a closed vessel in which an eddying motion of the mixture was produced by a fan. The degree of turbulence produced by the fan running at 2300 r.p.m. decreased the time of explosion from 0.14 sec. for the stagnant mixture of 0.03 sec., while the turbulence produced by a fan running at twice that speed effected only a small further decrease in the explosion time. Other factors dealt with were composition of charge, density, temperature, radiation from the hot cylinder walls, position of spark and shape of combustion chamber, and time of ignition.

EXPLOSIONS IN GASOLINE ENGINES

This paper, by H. T. Tizard, reviewed recent accurate experiments on that question, and the result of the review was that the specific heat of CO_2 at high temperatures was somewhat lower than that given by Bjerrum, that of water vapor was considerably higher, while the dissociation of CO_2 was about the same. The net effect was that the ideal efficiencies given by Pye and the author were, if anything, too high. There was nothing in those figures which explained why the thermal efficiency of the gasoline engine was as high as it was. A possible explanation was that the figures for dissociation, consistent though they appeared to be among themselves, were too high, and that either equilibrium was not reached at 2500 deg. cent. before the gases started to cool rapidly, or, more probably, that such large differences in temperature existed in the gas after explosion that it was inadmissible to calculate specific heats and dissociation on the assumption that the temperature was uniform.

The question of detonation in gasoline engines was also discussed, and reference was made to suggestions recently put forward by Callendar. (*The Engineer*, vol. 141, no. 3676, June 25, 1926. Report of the discussion on pp. 656-658. The papers abstracted here are on p. 658, *eg*)

Solid-Fuel Diesel-Type Engines

ACCORDING to the *The Times* (London) Trade and Engineering Supplement a German engineer, A. Schnürle, who has been working on the problem of solid-fuel engines, seeks to avoid the difficulties by effecting the combustion in a special chamber, which is in connection with the working cylinder but is separated from it by a porous filter-screen plate, through which gases can pass, but not coke and coal dust. The chamber, which is designed to withstand the working pressure of the engine and is well insulated, contains finely pulverized fuel which is in a glowing state, and as soon as fresh air is forced into it by the movement of the piston combustion begins and continues until the air is consumed. Then the piston reciprocates, and as a large volume of gas has been formed by the combustion of the fuel, the back pressure is much larger than that required to force the air into the combustion chamber.

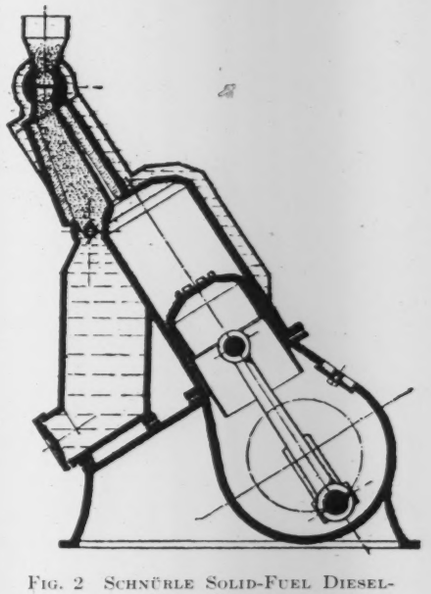

FIG. 2 SCHNÜRLE SOLID-FUEL DIESEL-TYPE ENGINE

Though the working principle is simple and difficulties with injection and with coal dust in the cylinder are avoided, the combustion chamber, which is under considerable pressure during the period of combustion, has to be of special construction to enable fuel to be admitted and ash removed. For this purpose small revolving valves are provided to lead the material to and from the stove at regular intervals, which may have a length of, say, half a minute. The ash falls into a container which is filled with water and is also under pressure. From this container it must be removed once a day. In order to promote regularity of burning the combustion chamber is set at an angle. Another difficulty is presented by the filter screen, which is always in a glowing state. In the trial engine it was made of zirconia, which has adequate strength and can stand temperatures up to 4200 deg. fahr. The screen is placed in a deep recess at the upper side of the combustion chamber, as shown in the figure. By this arrangement it becomes possible to give it a large surface offering little resistance to the gases which pass through it.

The thermal efficiency is naturally not as good as with the Diesel process. With a pressure of 25 atmos., however, the thermodynamic efficiency on the basis of indicated power is about 38 per cent, and with a mechanical efficiency of only 75 per cent the engine would burn only 0.93 lb. of 15,300-B.t.u. coal per hp-hr.

From the same article it appears that a large German engine works has again taken up the problem of the Diesel engine operating with coal dust, but formidable difficulties of various kinds have been encountered. (*The Times Trade and Engineering Supplement*, vol. 18, no. 418, July 10, 1926, pp. 375, 1 fig., *d*)

MACHINE PARTS

Heat-Treated Bolts

THE common bolt is such a familiar object that one is apt to overlook the fact that it violates many of the accepted rules for the design of highly stressed members. It starts with a sudden and sharp-cornered change of section under the head and finishes

with a closely pitched series of circumferential notches in the very place where the stress is highest. The most vital requirement in a bolt is a consistently high elastic limit to prevent stretching under exceptional loads coupled with a good elongation figure to accommodate a localization of stress in the sharp corners of the threads and the junction of the head with the shank. Ultimate strength is no guide to the stability of a bolt steel, nor is a high elastic limit anything but dangerous unless associated with the requisite ductility adequately distributed to meet local stresses.

The low elastic limit and uncertain physical properties of the common bolt have more far-reaching effects than is generally realized, and these are of a twofold nature. In the first place, a weak bolt stretches slightly under some momentary overload. If it forms the attachment of a part carrying varying or vibratory loads it now enters on a second stage of its career, during which it is subject to changing loads causing fatigue of the material. As long as it remains tight, holding the piece it attaches firmly to its mate, the load on it is substantially constant. If, however, it gets loose, the load varies from zero to a maximum, a condition which all engineers know to be much more harmful to the structure of the steel than a steady maximum load. In these circumstances fatigue of the metal sets in and the bolt may fracture under loads that appear to the designer to be well within the margin of safety. While the effects of insufficient strength are thus seen to be cumulative, the reverse is also true. A moderate increase in the strength of a bolt will not only prevent fracture, but also insure that the bolt remains tight.

On many bolts used as rigid fastenings and not as clearance-adjusting members for loose pieces, nut-locking devices should be superfluous. Their employment is a confession of the inadequacy of the common bolt to do its proper work in holding the pieces firmly together. All that they can do is to mitigate the effects of a slackness which should never have been allowed to develop. It is a very definite fact, demonstrated in many cases, that moderate increase in the strength of bolts will entirely stop the loosening of frame brackets and similar parts, which may seem at first to demand radical redesign.

Better bolts, then, would appear to be now essential. As regards material and methods of manufacture, it is a mistake to aim at a very high tensile strength in a bolt intended for universal employment. While it is of course possible to make bolts of special alloy steel having a strength of 80 or 90 tons per sq. in., such bolts would have no advantage in ordinary work over bolts of 45 to 50 tons tensile strength. A fitter using ordinary spanners would not be able to tighten them sufficiently to make use of the strength available. Further, many of the materials with which bolts are used would be crushed and distorted by the intensity of the bearing pressure of the nuts and bolt heads if the full strength of the bolt were employed. A bolt with an ultimate strength of 45 tons per sq. in. and a consistent elastic limit of 30 to 35 tons per sq. in., coupled with a fairly high percentage of elongation and a good Izod impact figure is excellent, and represents a great advance on the present-day commercial bolt.

Properly heat-treated after manufacture a good carbon steel is all that is required to give these results. If the head is upset from round bar just large enough to clean up for the shank any slight defects in the core of the billet from which the bar is rolled, it will not appreciably affect the strength of the finished article. The upsetting process also gives a much better molecular structure for the junction of the head with the shank. The grain of the metal flows round the corner into the head instead of passing straight through the head, as in the case of a bolt turned from the solid bar.

In order to relieve the strains caused by upsetting, the bolt should be normalized before machining, while the heat treatment should form the final operation. Scaling of the finished surfaces is prevented by maintaining a reducing atmosphere in the heat-treatment furnaces. Any slight discoloration that occurs is advantageous, enabling a heat-treated bolt to be distinguished instantly from an ordinary one. (Winning essay in a prize competition instituted by A. P. Newal & Co., Ltd., England, bolt manufacturers, by O. D. North. Abstracted through *The Automobile Engineer*, vol. 16, no. 217, July, 1926, pp. 252–253, *gp*)

POWER-PLANT ENGINEERING (See also Refrigeration: Water Treatment for Raw-Water Ice Plants; Testing and Measurements: Calculation of the Temperature of a Boiler Furnance)

German Boiler-Shell Practice

AN ARTICLE based largely on E. Höhn's book entitled Nieten und Schweissen der Dampfkessel (The Riveting and Welding of Boilers). This book gives full particulars from a number of tests made with various arrangements of straps, and illustrates alternative constructions. To avoid the use of continuous seams Höhn recommends the use of a number of short straps (Fig. 3) set across the seam to be strengthened. Such straps adhere so firmly that they break through the center before they can be drawn from the plate. Incidentally the author states that it is impracticable to transmit adequate stresses with riveted straps, but that

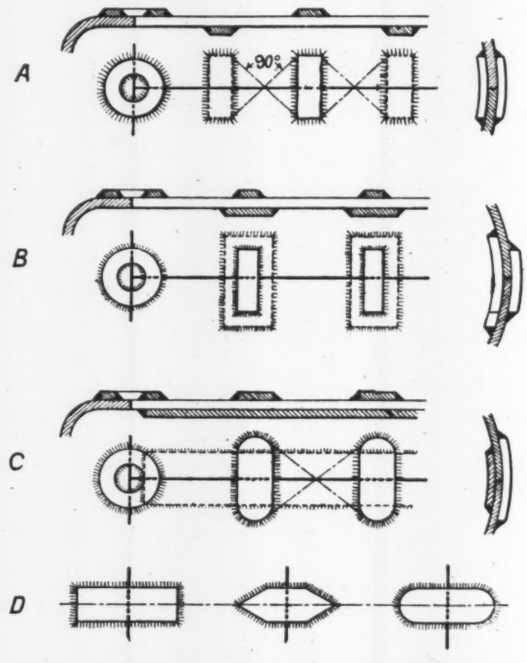

FIG. 3 TYPICAL SHAPES OF WELDED BOILER STRAPS

the welded strap is both technically and commercially well suited to the work.

Investigations concerning the elasticity of welded straps show that in all cases the strap connections are elastic, and when the strap is welded all around the variations in the stress at different sections are regular. The welds themselves participate in the elastic extension.

It being established that even thin square straps of the type considered break centrally, and not by tearing adrift, when welded all around, it is evident that such straps reinforce a seam—e.g., the longitudinal seam of a pressure vessel—by the full strength of the total section of straps. In order to eliminate the risk of the weld tearing before the strap breaks, the latter may be extended somewhat, perpendicular to the seam. Suitable dimensions are 3.9×2 in. (10×5 cm.), or 4.7×2.4 in. (12×6 cm.) for straps up to 0.47 in. (1.2 cm.) in thickness; and 5.9×2.4 in. (15×6 cm.), or 7.1×2.75 in. (18×7 cm.) for thicker straps. Typical arrangements for straps are shown in Fig. 3; that at (*A*) is suitable for plates up to $^5/_{16}$ or $^3/_8$ in. (8 to 9 mm.); (*B*) for any thickness; and (*C*) up to $^{25}/_{32}$ in. (20 mm.). Various shapes of straps are shown at the bottom of the figure. It is important that the reinforcing straps be not spaced too far apart, and Höhn recommends that they be so placed that lines tangential to the opposite ends of adjacent straps intersect at right angles, as shown in Fig. 3.

In case (*C*), Fig. 3, an internal strap running parallel to the seam is supplemented by external transverse straps, which counteract the bending moment set up by the internal strap, and also, by extending beyond the latter, compensate for the weakening of the

plate along the longitudinal welds. Due to the effect of error in bending the plate with and without contraction stresses in the longitudinal joint, the latter is nearly always subject to bending stress, but this can be compensated by the use of transverse straps. The strength of autogenously or electrically welded longitudinal seams is generally from 65 to 80 per cent that of the plate itself, but by adding transverse straps (welded all around) of a total section equal to 35 per cent of the longitudinal section of the plate, the resultant strength of the seam equals or exceeds that of the plate. This claim is proved by the fact that vessels with such reinforced seams burst in the solid plate when tested to destruction.

At the Mulheim Works of A. Thyssen & Co., welded drums are being made up to 4 ft. 1 in. in internal diameter and 31 ft. long. These drums are rolled all in one plate, including the drum ends, i.e., with one longitudinal seam and no circumferential seams.

The edges of the plate are machined before welding, which helps to insure a homogeneous and close weld. It is also very necessary, as the strain set up in the plates during shearing is considerable and extends some 5 or 6 in. from the edge of the plates, the latter being 2 in. thick. After machining, the plates are annealed at a temperature of 910 deg. cent.

After water-gas welding, the cylinder is again annealed to remove all strains set up during the welding process; from the annealing furnace and while red hot, the cylinder is threaded on a bending-roll mandrel, where it receives its truly cylindrical shape. It is kept revolving on the rolls while cooling, thus avoiding distortion. The ends are then trimmed off and the shell tested to 1.5 times the working pressure as a preliminary test; if this is withstood satisfactorily, the manufacture is proceeded with.

The hemispherical ends are now formed under pressure, leaving an opening for manholes. During this operation the thickness of the metal increases naturally, strengthening the end around the manhole opening. The final operation consists in machining the drum where required.

The Krupp Co., at its Essen Works, makes solid-forged boiler shells and solid-forged drums with closed-in ends. These hollow forgings are supplied either rough or machined. The drums are finished with either an oval manhole at each end or an oval manhole at one end and a round hole at the other. The diameter of this round hole depends upon the size of the drum. For pressures up to 500 lb. per sq. in. plain carbon steels are used, and for higher pressures, and especially high temperatures, two grades of nickel steel are used, details of which are given in the original article.

Among other things, it is stated that the ordinary formula for calculating the thickness of boiler plates cannot be relied upon for high-pressure work. (W. S. Findlay in *Power Engineer*, vol. 21, no. 244, July, 1926, pp. 245–248, 3 figs., *g*)

An Unusual Steam Turbine

THE General Electric Company is now building a 3600-r.p.m., 1875-kva., 480-volt, 3-phase, 60-cycle turbo-generator set to meet the requirements of the American Trona Corporation whose plant is located near a large dry salt lake in the Mohave Desert of California.

Large quantities of steam are used for triple-effect evaporators in the process of separating the salt into its constituent parts—borax, potash, and other chemicals. The new turbine will receive saturated steam at 150 lb. and operate against a back pressure of 28 lb. kept constant by a regulator control acting through the governor synchronizing equipment.

Exhaust steam is fed to a common header which receives the exhaust of the steam-driven pumps and other equipment as well. The turbine therefore acts as a reducing valve and at the same time obtains from the process steam the available power between boiler and process pressures. Additional power required by the plant is furnished by the Southern Sierras Power Company from their hydroelectric system. Formerly the power company supplied all the power used, so the new unit should effect a considerable saving.

As the American Trona Corporation's equipment will be in synchronism with that of the power company, the sudden load might cause dangerous overspeeding of the turbine, and to prevent this the emergency governor-tripping mechanism will be equipped with a cut-out switch to throw the generator off the incoming power lines before dangerous overspeeds can develop.

While the turbine is designed to give best economy during normal operation at 28 lb. gage back pressure, it will also operate at high economy during emergency condenser operation, thus conserving water—the most important consideration of the development. Mountain spring water will be used in the surface air cooler which cools the generator, and this water, after passing through the cooler, will be fed to the boiler-feedwater heaters and thence into the boilers. Thus the generator losses absorbed by the cooling air in the form of heat will be transmitted to the cooling water and conserved. (*Power Plant Engineering*, vol. 30, no. 13, July 1, 1926, p. 734, *d*)

Doubling Boiler Capacity

AT THE Fordson power plant of the Ford Motor Co., on the River Rouge, an important problem came up some time ago. The plant, originally laid out for 65,000 kw., was increased first to 100,000 and then to 240,000 kw. capacity. In the boiler room there were eight units and practically no space for any more. These units were designed for continuous running at 250 to 300 per cent of nominal rating when burning simultaneously blast-furnace gas and powdered fuel. The problem of how to make the furnaces do added duty was solved in the following manner.

On the first furnace to be remodeled the number of powdered-coal burners was increased from six to twelve on each side, giving a coal-burning capacity of 72,000 lb. per hr., and to permit operation up to 660 per cent of rating, water cooling for the side walls and a water screen across the bottom of the furnace were provided. Forced- and induced-draft and primary air fans of adequate size, plate-type air preheaters, and an elaborate system of controls were added features in the furnace-rebuilding program. The work on one furnace has been completed recently, and the installation is being given a thorough trial before the remodeling of the other seven is undertaken. Up to the time of this writing, the unit had been running continuously for three weeks, 24 hr. a day, at 500 per cent of the original rating of the boiler.

An unusual feature is the arrangement of the side-wall cooling surface and the water screen at the bottom of the furnace into complete circulating systems independent of the boiler proper, in which steam is generated, passed through radiant superheaters behind the side-wall tubes, and delivered under the same conditions, 225 lb. gage pressure and 200 deg. fahr., as the steam from the boiler.

The radiant superheater was installed so that it is protected from excessive temperatures by the side-wall surface and at the same time, owing to the omission of the fins on the side wall, tubes at this point may absorb the radiant heat from the fire. The convection superheaters in the first bank of boiler tubes on either side were retained and the radiant superheaters installed in addition, as it would have been impossible to add sufficient convection surface to take care of the increased output of steam without undue pressure drop.

In addition to the water-cooling surface previously mentioned, it was considered desirable to protect the arches guarding the lower boiler drums. In front of each of the two arches are thirty-four $3^1/_4$-in. tubes spaced on $7^1/_2$-in. centers, which connect into a header at their lower ends and into the boiler drum at their upper ends. Loose-fit tubes continue through the drum into the last row of tubes in the center bank, so that the circulation is positively directed toward the upper boiler drums. The headers are supplied with water from the lower boiler drums through four 4-in. tubes at both ends of the drums. In the two arch screens there is 750 sq. ft. of heating surface, this being distinct, however, from the other cooling surface, as it is connected directly into the boiler circulation.

Air preheaters are of the plate type and are enclosed in a heavy casing, outside of which is 3 in. of insulation covered by light-gage panel plates. In the preheaters the air is raised in temperature to a maximum of 519 deg. fahr.

Air is supplied to the feeders at a temperature of 200 deg. fahr., which means that under certain conditions such as at maximum rating it was necessary to lower the temperature of this air before it entered the feeders, of which there are six at each side of the boiler, each driven by an individual motor capable of handling

6000 lb. of coal an hour. To do this cooling, a specially designed damper box was provided at the intake of the fan which has one side open to permit drawing in air at room temperature. At the side of this box there is connected to the damper shaft a thermostat control with a spring balance regulating the amount of cool air drawn into the fan and keeping the temperature at approximately 200 deg. fahr.

By these methods the amount of coal that can be burned per boiler has been raised to 72,000 lb. per hr. and the boilers made to operate at over 600 per cent of rating. (*Power*, vol. 64, no. 2, July 13, 1926, pp. 40–44, 7 figs., *d*. Compare editorial on p. 57)

POWER GENERATION

Electricity Direct from Coal

IN AN ARTICLE recently published, Ulick R. Evans discusses the production of electricity direct from coal by means of some form of primary cell. In the primary cell electricity is generated by the chemical action of the component parts of the cell, one part, for example, zinc, being the consumable element. Many attempts have been made to use coal or some substance derived from coal as the combustible element in such a cell, and to this the name of "fuel cell" has been given. Hitherto, however, all such cells while developing electricity in small quantities and at a pressure of less than one volt have suffered from the disadvantages that they rapidly polarize, which means that the carbon becomes coated with bubbles of gas and the action dwindles until it finally ceases.

In view of this great drawback of the fuel cell, the metallurgical fuel cell has been devised in which, as the name implies, metal poles are employed and various gases are utilized to create the electrochemical potential difference in virtue of which the cell yields an electrical current. The author gives a description of some of these and indicates how by the use of carbon monoxide or water gas, and possibly of producer gas manufactured from coal or some other fuel, electricity can be said to be obtained directly from fuel. So far, the degree of success obtained is not impressive, and, in particular, the size of the cell in relation to its output is excessive, so that the possibility of using a battery of such cells for commercial purposes at the present time is rather remote. However, the germ is there, and it may be capable of immense development. It is claimed that there is no fundamental reason why an efficiency of 95 per cent could not be obtained if a satisfactory cell could be developed, and meanwhile, expanding the ideas enumerated above, it is suggested that an attempt might be made to design a practical form of battery of, say, ten cells fed by coal gas which could be used for charging accumulators. The practical experience thus gained would make it possible to decide whether there was reasonable hope of developing the system still further for use in the generation of electrical power on a large scale. (*Royal Technical College, Glasgow, Metallurgical Club Journal*. Abstracted through editorial in *Colliery Engineering*, vol. 3, no. 28, June, 1926, pp. 242–243, *g*)

RAILROAD ENGINEERING

The Development of the Turbo-Condensing Locomotive

Because of the fact that the condenser is the least developed part of the turbo-condenser locomotive, the author discusses this problem first.

For practical purposes the possible substances which may be employed to carry away the latent heat which is liberated when steam is condensed may be either air or water, or a mixture of both.

Fig. 4 gives four curves, the vertical ordinates of which represent British thermal units and the horizontal ones temperatures in degrees fahrenheit. It is supposed that 100 lb. of air is kept in a saturated state from freezing point right throughout the range of temperatures shown. As the temperature rises the ratio of weight of water vapor to that of air gets greater at a rapidly increasing rate. Curve A represents the heat absorbed in raising the temperature of the water to that of vaporization. Curve B shows the heat contained in 100 lb. of air, and curve C gives the amount of heat spent in vaporizing the water. Curve D represents the sum of curves A, B, and C, and therefore gives the total heat of evaporation.

It will be seen that the part played by vaporization in absorbing heat is much greater than that of heating the air, and at the higher temperatures the difference increases rapidly. It follows, therefore, that if the evaporative principle be used in condensing steam, a far less volume of air will be required than if air be used alone. As both the horsepower necessary for the supply of air and the amount of cooling surface are roughly directly proportional to the quantity of air, it is obvious that if water is obtainable as a condensing medium it is much more economical to make use of the evaporative principle, even if the quantity be so limited that only partial saturation can be achieved. The question is purely one of economy, and the selection should therefore be made of the system which gives the smallest initial cost and subsequent upkeep.

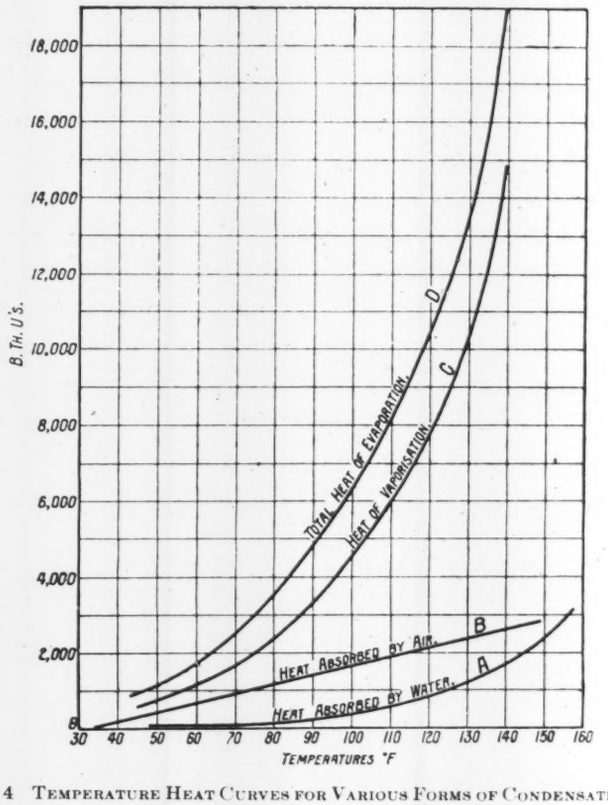

FIG. 4 TEMPERATURE HEAT CURVES FOR VARIOUS FORMS OF CONDENSATION

The property of the rapidly increasing steepness of the curve C makes the evaporative principle specially adaptable to hot climates, because for small rises in temperature large quantities of heat are absorbed by the condensing medium.

In the arrangement preferred by the author the condenser is coupled to the locomotive in front of it, as this makes the exhaust pipe and other connections between the engine and condenser short and simple.

The process of condensing consists in producing a continuous thin film of water on the outsides of the tubes while the exhaust steam passes through them. At the same time, air at a high velocity is passed over the film. Very rapid evaporation of the water surrounding the tubes takes place, and since the heat required for this is supplied by the steam, correspondingly rapid condensation of the steam is the result. Since water is a bad conductor of heat, it is essential that the film be kept as thin as is physically possible in order that the resistance to the flow of heat be a minimum. At the same time the surfaces must not be allowed to become dry if the maximum efficiency is to be preserved, because any dry portions simply become air-cooled, and a glance at the curves of Fig. 4 shows how rapidly the transfer of the heat will drop. It will be seen that to secure efficient filming calls for careful design and arrangements for adjustment.

Mention has already been made of the importance of turbulence of the air in its passage through the condenser. Provided that the water is introduced in the correct way and at the proper places in the condenser and that the correct balance between air and water

is maintained, the turbulence assists in no small degree in the attainment of ideal filming, hence it serves a double purpose.

Regarding the balance between the air and water, or the ratio between the quantities of air and water, it may appear that condensation would increase with the quantity of air pumped over the surfaces. Up to a certain point this is true, but beyond it it is not so.

To reiterate the property of the curve D of Fig. 4, the heat absorbed by the condensing medium increases rapidly as the temperature rises, so, obviously, the more the air is heated in transit the more is the economical use made of it. The final temperature of the air is limited principally by the temperature at which condensation takes place or, what is the same thing, by the pressure of the exhaust steam. A certain final temperature difference has to be allowed or there would be no transfer of heat through the walls of the ultimate part of the tubes, and it is the difference which determines the amount of air which is required. In each set of conditions there is one final temperature difference which gives the greatest overall economy, and the predetermination of it is one of the various operations which has to be carried out by a process of trial and adjustment. If the proper quantity of air is exceeded the temperature will drop, and this will be accompanied by a fall in the heat content per pound of air and no beneficial effects will result by the increase. If the increase be made to any considerable extent, the power of the air for absorbing vapor will be so weakened that a decided decrease in condensation will follow.

Details of the construction of the condenser, and especially the fan, as well as the flexible connection between locomotive and condenser, are described in the original article. (First of a series of articles by George F. Jones in *The Railway Engineer*, vol. 47, no. 558, July, 1926, pp. 233–238, 5 figs., *gd*)

REFRACTORIES (See Engineering Materials: The Mechanism of Spalling)

REFRIGERATION

Water Treatment for Raw-Water Ice Plants

IT IS ONLY in soft-water districts that untreated water makes first-quality ice, and even then only when the water is of such a character that the suspended matter can be removed by filtration and no other deleterious ingredients are present.

It is therefore important to determine the character of water and its suitability for ice making before the machinery is selected and installed. The objectionable impurities to be found in various quantities in nearly all water supplies are calcium, magnesium, and sodium salts, iron, alumina, suspended matter, and organic matter. Of the above the carbonates and bicarbonates of calcium and magnesium, the sulphates and chlorides of magnesium, the iron, and the organic matter are the most objectionable, because when they are present even in small quantities they are the cause of objectionable deposits in the ice which may range in color anywhere from gray to dark brown, depending on the kind and amount present.

Treatment of the water with hydrated lime in an efficient water-treating plant will practically eliminate the calcium and magnesium carbonates, will entirely eliminate the iron, will greatly reduce the alumina and silica, and, followed by filtration, will eliminate the suspended matter and organic matter. The use of soda ash to soften the water is rarely found in the treatment of water for the manufacture of raw-water ice. Use of treated water is said to reduce heat consumption in making ice, and the author gives a calculation to prove it which would show that water treatment means a saving in refrigeration of about 8 per cent.

Zeolite treatment does not seem to be applicable in raw-water ice making. It is true that it gives a water of zero hardness, but it replaces the calcium and magnesium salts with sodium salts and therefore does not really reduce the mineral content of the water; ice made from water so treated will have heavy opaque cores, and oftentimes heavy opaque shells around the lower side and bottom of the cake. Zeolite treatment also has no effect on the iron, and even a trace of the latter gives objectionable color to the ice. (W. N. Waterman in *Power Plant Engineering*, vol. 30, no. 13, July 1, 1926, pp. 758–760, *p*)

In this connection another paper, by A. S. Behrman, is of interest. After discussing briefly the general subject of water hardness and its influence on ice making, the author suggests the following test as a quick, rough method of determining water hardness, capable, however, of giving results sufficiently accurate for general purposes.

The method suggested is the "soap test," and consists simply in finding out how much soap a given water requires to form a lather. If a measured sample of the water is taken in a bottle and a soap solution of a certain definite strength added to it a little at a time, shaking the bottle after each addition until a lather is formed, the hardness of water can be calculated from the amount of soap solution used. (Paper by A. S. Behrman of the International Filter Co., read before the convention of the N.A.R.E.; abstracted through *Power House*, vol. 19, no. 12, June 20, 1926, p. 29, *p*)

SPECIAL MACHINERY

A New Rotary Drilling Machine

THE machine described here combines certain features of the diamond drill and standard rotary. It is built in the United States by the Sheldon Burden Drilling Co. In its design the engineers have combined the hydraulic-feed principle of the diamond drill with the rotary table, and this has been done in such a way that

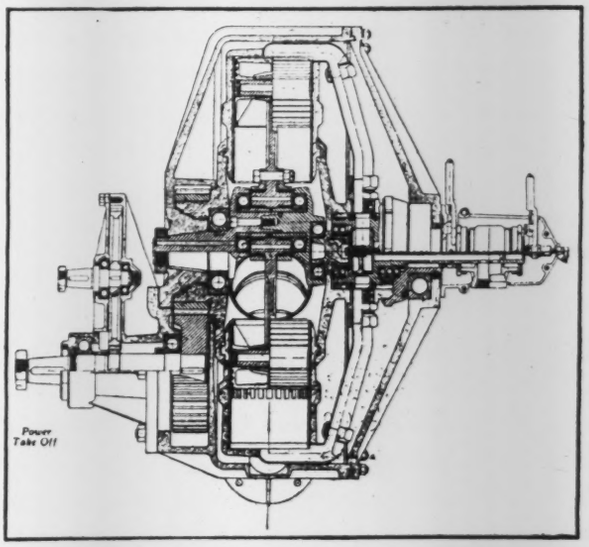

FIG. 5 CROSS-SECTION OF ROTOBAKER STEAM MOTOR

the drill stem can be operated from the regular feed of the drum as in regular rotary practice, or hydraulic feed can be put into service.

Since the weight of the pipe is on the hydraulic cylinder, deeper drilling is assured without the hazards from twisting. As the condition of the strata varies, the feed can be increased or decreased by valves on the control side of the machine, which either let water out of the cylinder or into it. The speed of the machine can be varied over a great range. The hydraulic feed is an obvious advantage in the coring of hard formations with diamond bits or in soft formations, using one of a number of double-tube coring barrels which are made for rotary drilling.

The power for the machine is developed by a Rotobaker steam motor, Fig. 5, having a rating of 100 hp. and a range of speed up to 900 r.p.m.

Friction clutches, properly engineered, connect the motor to both high and low drum speeds, as well as to the rotary drive. The hydraulic cylinder is rigidly attached to the rotary table frame below the rotary, while the hydraulic piston head and movable parts travel for a 24-in. stroke through the inside of the table, and can exert a pressure of about 60 tons, either up or down. The flush-point pipe is gripped by a patented device that rotates and holds the pipe at all pressures, and the drilling is accomplished through a stuffing box. In soft formations, at shallow depths, the rotary drives the regular grief stem by means of regulation drive bushings,

and the hydraulic-feed attachment can be put into place or removed by any driller in a couple of hours. The machine with its hydraulic device is mounted on a 12-ft. frame securely anchored by three U-bolts around the steel base of the derrick. Its total weight is about 10 tons. If required the motor can be removed and either electric or gasoline power substituted.

In hard formations ordinarily considered as good cable tool ground the machine has, the writer was informed, done 600 ft. in 24 hours. (H. R. Berry in *The Petroleum Times*, vol. 15, no. 387 (old series 1246) June 12, 1926, pp. 953–954, 3 figs., *d*)

The Pantagraver

THIS is a new instrument developed by the Naval Hydrographic Bureau for the preparation of nautical charts. The basis of this instrument is an adaptation of the pantagraph used by mechanical draftsmen. Data from survey sheets, maps, and charts of various scales are transferred directly to the copper plate and compiled into a finished map or chart of one standard scale.

The Navy has built the only instruments of this kind in use. They were all constructed at the Naval Gun Factory of the Washington Navy Yard from designs prepared by the inventors of the apparatus, John H. Larrabee and T. Peter Lampe. Two are operated by the Hydrographic Office, and a third has just been installed in the Engineering Reproduction Plant of the Army.

The first pantagraver was made of steel, but proved to be heavy and did not respond as readily as desired. Others were built of duralumin and are lighter and easier to operate. (*Daily Metal Trade*, vol. 16, no. 135, July 9, 1926, p. 1, *d*)

SPECIAL PROCESSES

Chrome Plating

FROM the article under consideration it would appear that while chrome plating has been successfully applied recently on a commercial scale, it was not until 1919 that a German patent overcame most of the faults of earlier processes; but even this last process is subject to very great fluctuations due in part to impurities found in the chemicals purchased in bulk, and in part to the "personal equation" of the workmen. The author therefore recommends that small platers should wait until both the technique of the process and the patent situation clear up. At present, he states,

> We have several of the country's most powerful corporations plunging into the chromium-plating argument, ready for a fight to the bitter end. To name them all would be like calling the roster of the leading automobile companies, electric corporations, sewing-machine and phonograph companies, and a long list of firms affiliated or connected with them as manufacturers of accessories.
>
> From present indications it appears as if the main battle will be between the Metal and Thermit Corporation, the Chemical Treatment Company, the General Motors Corporation, and the General Electric Company, with more than a hundred associated corporations either actively connected with and engaged in the fight or watching the coming struggle from the side lines. The fight has not yet started, the first gun has not yet been fired. But preparations for the battle are going on, and lawyers are reading up on chromium plating who never knew such a process existed. The electroplating world will witness a struggle that promises to be as bitter and as far-reaching as the famous Selden patent struggle was in the automobile industry some sixteen years ago. And the small plater really cannot do better than await the outcome of the big fight, before committing himself to the use of either of the patented processes.
>
> The patents already issued, and the score or more patents of extremely broad claims which are now pending in the U. S. Patent Office, cover almost every conceivable mixture of strength of bath, voltage and amperage of current used, and additions of the most likely salts to the bath. In order to find a solution and process not covered by these patents an investigator will have to devote several years to research and trial work, and this phase brings the recent work of the chemists of the General Motors Company, the General Electric Company, and other manufacturing concerns into direct competition with the chemical concerns who were first in the chromium-plating field.

(Theodore R. Von Keller in *Brass World*, vol. 22, no. 6, June, 1926, pp. 181–183, *g*)

STEAM ENGINEERING

Notes on the Comparison of Steam-Turbine Efficiencies

BECAUSE of lack of space only certain portions of this interesting article can be reported here. The fuel consumptions chargeable to steam-turbine operation vary so widely with steam conditions and the different heat cycles that it is often hard to determine the relative thermal advantage of each of several arrangements. This situation is due in part to the difficulty of getting a true comparison. On the one hand there is the thermal efficiency (low on the percentage scale), in which the operator is interested because it directly affects his fuel consumption. On the other hand there is the stage efficiency (high on the percentage scale), which is the index of perfection of the intrinsic design of the unit and in which the designer is interested. In addition to these two efficiencies there are also the turbine efficiency, the cycle efficiency, and the reheat factor, all equally important and all interrelated and also dependent on the steam conditions.

There are certain fundamental relations between stage efficiency, reheat factor, turbine efficiency, cycle efficiency, and thermal efficiency which by means of certain definitions may be confined wholly to the properties of expanding steam. These relations have been discussed in the literature and usually give the impression that the subject is very complicated. But the charting of the various values shows that the relations become simple and easy to handle both

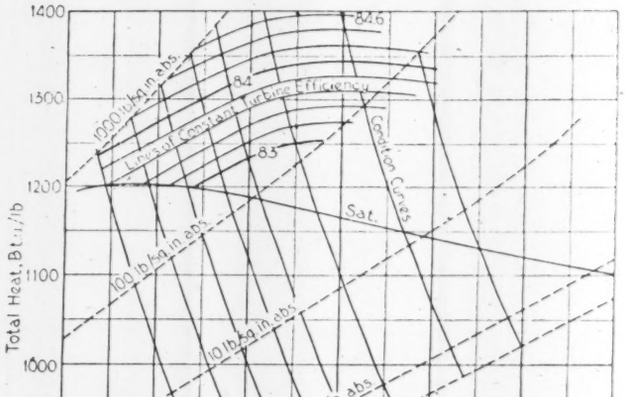

FIG. 6 TURBINE EFFICIENCIES FOR VARIOUS INITIAL STEAM CONDITIONS AND EXPANSION AT 80 PER CENT CONSTANT STAGE EFFICIENCY TO 1 IN. HG. ABS. BACK PRESSURE

for the Rankine cycle and for the extraction cycle. It is the purpose of this article to show how the supposition of an equivalent uniform stage efficiency permits its quick determination under any particular set of conditions.

Since the term "turbine efficiency" is ordinarily used in a "throttle-to-switchboard" sense, it is necessary to call attention to the more restricted meaning which is used in the present article. It might be called a "nozzle-entrance-to-shaft" efficiency since generator and bearing losses are counted in as power output while throttle, high-pressure-packing, and leaving-velocity losses are omitted from the theoretical power. This restriction is possible because each of the five types of loss noted can usually be treated as a lump figure independent of all others. Turbine efficiency is therefore the ratio of the power output at the shaft to that theoretically available from the flow in the blade system and interstage packings, between the pressure and superheat at the first nozzle entrance and the pressure at which it finally leaves the blade system.

The other terms used have their usual meanings restricted only by the limited viewpoint just explained. Thus steam conditions refer to first-nozzle entrance and last-blade exit, while a statement of feedwater temperature must be included if the extraction cycle is not completed by heating to boiler temperature. Cycle efficiency refers to the ratio of the heat theoretically available for useful conversion to the net heat supplied in the steam, and thermal efficiency is the product of turbine and cycle efficiencies, giving the ratio of the power output at the shaft to the net heat supply, due credit being given for throttle, high-pressure-packing, and leaving losses.

In defining stage efficiency and reheat factor it is necessary to bear in mind that the summation of the available adiabatic heat drops in the individual stages always exceeds the overall adiabatic heat drop. This is because the energy lost in friction and eddies in each stage goes at once into heat and is absorbed by the steam, so that it is available as heat energy for partial recovery in all the stages that follow. The reheat factor is the ratio of the summation of stage heat drops to the overall adiabatic heat drop, and since the overall stage efficiency is computed relative to the overall heat drop, and since the overall stage efficiency is computed relative to the overall heat drop, it follows that the ratio of turbine efficiency to stage efficiency is also the reheat factor.

Fig. 6 is a total-heat-entropy diagram from which most of the usual pressure, quality, and temperature lines of the Mollier chart have been left out, but the condition of the steam at any stage of the turbine is shown just as well by its total heat and entropy as by its pressure and temperature or quality. Condition curves for expansion at uniform stage efficiency have been drawn. Each of these curves represents the steam during an expansion in which there is a fixed percentage of the adiabatic available energy lost in each infinitesimal pressure drop. This fixed percentage loss of the available heat appears in reheat, slightly increasing the entropy and causing the expansion curve to slope away from the adiabatic in the direction of increasing entropy. The slope of a condition curve for any particular efficiency depends only on the slope of the pressure curve at each point in the expansion, and each of these condition curves shown in the diagrams was first laid out upon a regular Mollier chart containing all of the pressure lines.

The slope of the pressure curve shows the heat added per unit increase of entropy. The slope of the condition curve shows the useful heat converted per unit increase of entropy. Hence the slope of the condition curve bears the same ratio to the slope of the pressure curve as the useful fraction lost in friction and appearing in reheat. Thus if the constant stage efficiency for an infinite number of stage is e, the ratio of slopes is $e\,(1-e)$, and this simple relation enables the graphic construction to proceed from pressure to pressure until the whole condition curve has been drawn.

Fig. 6 gives curves for 80 per cent stage efficiency and another figure in the original article (Fig. 2) gives curves for 70 per cent stage efficiency. In every case the reheat in the early stages of expansion is partially available for conversion in the lower stages. The turbine efficiency thus always exceeds the stage efficiency, and this ratio, which is greater than unity, is called the reheat factor. The condition curves enable the immediate scaling of the total useful heat for expansion between any initial condition and any exhaust pressure. Comparison with the available adiabatic heat gives the turbine efficiency, which may be divided by the stage efficiency to give the reheat factor without reference to formulas. This is illustrated by an example in the original article.

It is usually true that turbines having the same stage efficiencies show higher turbine efficiencies with improved steam conditions, that is, higher initial pressures or superheats. Thus Fig. 6 shows that a turbine with 80 per cent stage efficiency operating at 150 lb. per sq. in. abs. and 90 deg. fahr. superheat would have a turbine efficiency of 83.0 per cent, while another machine with the same stage efficiency but operating at 400 lb. per sq. in. abs. and 200 deg. fahr. superheat would have a higher turbine efficiency, namely, 84.0 per cent.

While Fig. 6 and Fig. 2 of the original article show plainly how higher steam conditions give higher turbine efficiencies for stage efficiencies of 80 and 70 per cent, due to greater benefit from reheat, other diagrams would be desirable to illustrate the possible turbine efficiencies with intermediate stage efficiencies as well as values above 80 per cent. Such diagrams would also make possible the determination of stage efficiencies corresponding to given turbine efficiencies.

Fig. 7 takes care of these questions by showing reheat factors for a number of initial pressures and temperatures plotted against such efficiency in all cases for an exhaust pressure of 1 in. of mercury. In the original article curves are drawn for six different pressures. Only two of these are here reproduced.

Thus with 500 lb. per sq. in. initial pressure and 700 deg. fahr. temperature, with a stage efficiency of 84 per cent the reheat factor would be 1.04, giving a turbine efficiency of 87.3 per cent. On the other hand, with a known turbine efficiency of 85 per cent under the same steam conditions, one or two trials show that the reheat factor must be 1.05, giving a stage efficiency of 81.0 per cent. Variation of the reheat factor with pressure is very small, so that reheat-factor curves for intermediate pressures may be easily found by interpolation between the diagrams.

If two machines show the same turbine efficiency while the steam conditions are different, it is usually true that the turbine with the poorer initial conditions, that is, lower initial pressure or superheat, will have the higher stage efficiency, indicating that it is more perfect from a design point of view. Thus, one or two trials on Fig. 7 show that a machine with a turbine efficiency of 83 per cent at 200 lb. per sq. in. abs. pressure and 400 deg. fahr. temperature would have a stage efficiency of 80.0 per cent, while another machine with the same turbine efficiency operating at 500 lb. per sq. in. abs. and 700 deg. fahr. would have a stage efficiency of only 78.2 per cent. Thus if the more perfect mechanical design with the 80 per cent stage efficiency were adapted, without impairment of its stage efficiency, to the better steam conditions of 500 lb. per sq. in. and 700 deg. fahr. temperature, it would show a turbine efficiency of 84.4 per cent and would thus be preferable both from the design point of view and from the point of view of thermal economy. Again, attention is called to the non-thermodynamic effects. In the example cited the moisture in the lower stages has been much reduced, making the efficiency easier of attainment, but on the other hand the pressure has been increased, making it more difficult to preserve the 80 per cent stage efficiency during the change of conditions. These influences must be allowed for by methods which are applicable only to the particular type of blade system under consideration.

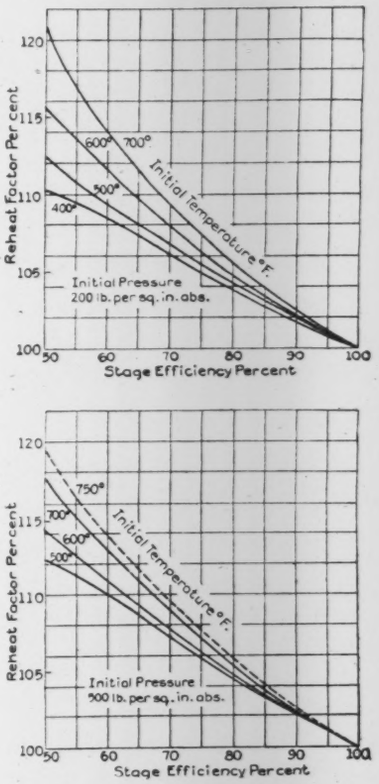

FIG. 7 REHEAT FACTORS FOR STEAM EXPANSIONS WITH CONSTANT STAGE EFFICIENCY AND INFINITE NUMBER OF STAGES TO 1 IN. HG. ABS. BACK PRESSURE

The author discusses next the amount of heat recovery and the extraction cycle represented by feedwater heaters. This part of the article cannot be abstracted because of lack of space. (Ernest L. Robinson, Mem. A.S.M.E., Turbine Engineering Dept., General Electric Co., Schenectady, N. Y., in *General Electric Review*, vol. 29, no. 7, July, 1926, pp. 503–510, 10 figs., *tA*)

The Limiting Efficiency of the Reaction Steam Turbine

AFTER a consideration of the figures obtained from Professor Stodola's tests on Brown-Boveri blading (which gave an actual hydraulic efficiency of 90 per cent and an efficiency of 96 per cent after correction for tip leakage) and Dr. Maguerre's investigation on the kinetic energy carried over from one row of blades to the next, the author comes to the conclusion that it is clearly important to use effectively the "carry over" from one row of blades to the next.

This appears to be getting more and more clearly recognized by designers, who now frequently use conical casings in which the blades vary in height continuously from row to row in place of being split up into a series of groups of blades with abrupt changes in the blade height from group to group. The importance of avoiding these abrupt changes in blade heights is, as already indicated,

enhanced by the present trend toward higher and higher rates of revolution. With the standard arrangement of groups this implies fewer and fewer rows per group, and the loss by "carry over" at the last row of the group is thus spread over a smaller total of stages, which must involve a diminution of the efficiency.

Conical groups of blades seem to be specially desirable in the low-pressure turbine, where we have nowadays a combination of large diameters with high rates of revolution. The group therefore often consists of a couple of rows only. Although the steps in blade height are then generally not large, relatively speaking, each of these sudden increases in the size of the steam way must tend to diminish the efficiency with which the "carry over" is utilized. If all of it were wasted the "indicated" hydraulic efficiency of such two-row groups could not, as shown above, exceed some 87 per cent, even if tip leakage could be entirely prevented. As this is impossible the actual figure would be materially less. Experience seems to show that a fair efficiency is attained even with this two-row arrangement. This would seem to imply that a considerable fraction of the "carry over" is effectively utilized in the succeeding group. It would appear, nevertheless, that better results should be attained by restricting this division into groups, the necessary increase in the steam way being provided for by varying the blade height continuously from row to row. Professor Stodola's tests were made with such a group, which contained in all no less than 56 rows of blades. (*Engineering*, vol. 122, no. 3155, July 2, 1926, p. 1, *tg*)

TESTING AND MEASUREMENTS

Calculation of the Temperature of a Boiler Furnace

IN ORDER to calculate the furnace temperature the author compares a boiler furnace with a room, mostly filled with radiating gas, provided with a door through which the gas exits to the boiler, and in most cases also with a window, which represents the heating surface of the boiler exposed to radiation from the furnace. The theoretical temperature depends only upon the composition of the fuel and upon the amount and temperature of the combustion air. Calculations based on the theoretical temperature do not, however, represent actual conditions because there is always a certain amount of radiation to be considered, and also dissociation is present in the furnace. The original article gives a diagram showing the radiation to the exposed surface of a boiler at varying temperatures, or the radiation in a unit boiler with 1 sq. ft. of window surface.

For a given input of heat and for a given theoretical temperature the real furnace temperature is found in the diagram by drawing a straight line from corresponding points of the axes of the diagram. The point sought is the intersection between that line and the curve of radiation.

In this manner the diagram, Fig. 8, is graphically calculated. Its upper part gives the real furnace temperature at various conditions, while the lower part indicates the efficiency of the furnace window, i.e., the percentage of heat input directly absorbed by the window through radiation.

The method of calculation is, however, not mathematically correct. It presumes that the specific heat of the fuel gas (C_p) is constant at varying temperature, while in reality C_p increases with the temperature. The difficulties are overcome, however, if the C_p that is correct for the furnace temperature is also used in the calculation of the theoretical temperature.

Moreover, the variation of C_p for different furnace temperatures and for different compositions of coal gas is insignificant. The figure $C_p = 0.275$, calculated for a furnace temperature of about 2400 deg. fahr., will be fairly accurate for all practical cases.

The theoretical temperature is found by dividing the heating value of the fuel by C_p and by the weight of fuel gas per pound of fuel.

To be able to use Fig. 8 for the calculation of the temperature of a furnace, it is necessary to estimate the area of the furnace window. This is easily done where the furnace is provided with a single window as is the case with most of the boilers, but it will cause some trouble in furnaces completely surrounded by water-cooled walls. In this case it is not likely that all corners of the furnace are filled with radiating gas. At all events, the gas in a corner will obtain a lower temperature than the gas in the middle of the furnace. For such boilers one has to calculate with a reduced window surface, found by experience.

If the furnace is contracted between grate and boiler, the smallest area is to be regarded as window when calculating the temperature of the lower part of the furnace.

An example will show the use of Fig. 8. In a boiler with a window of 100 sq. ft. 3000 lb. of coal with a heating value of 12,000 B.t.u. per lb. is burned hourly. The CO_2 content in the furnace is assumed to be 14 per cent.

The input per square foot of window is $3000 \times 12{,}000/100 = 3.6 \times 10^5$ B.t.u. The theoretical temperature, read from the table, being 3350 deg. fahr., the furnace temperature is found in the diagram at the point where a curve for 3350 deg. fahr. cuts a

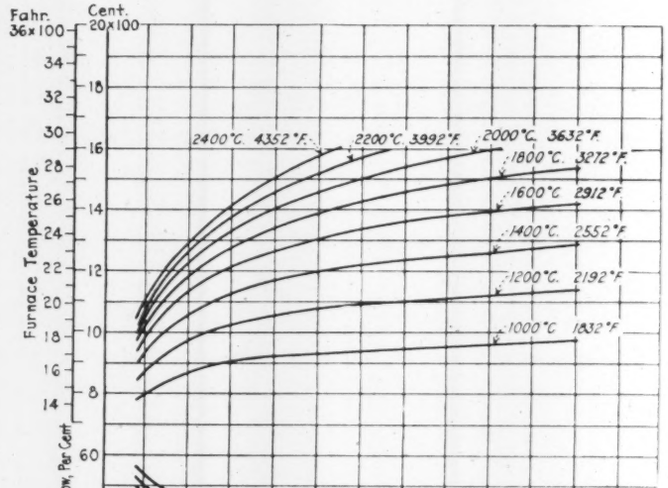

FIG. 8 DIAGRAM FOR CALCULATING TEMPERATURES OF A BOILER FURNACE

vertical line through 3.6×10^5, or slightly above 2400 deg. fahr. The corresponding efficiency of the furnace window is found in the lower part of the diagram to be about 24 per cent.

The diagram may also be used to find the maximum load on a boiler when the surface of the window and the highest allowable temperature are known. The influence of preheated air upon the furnace temperature may easily be followed in the diagram, the rise of furnace temperature for a given air temperature being low in a furnace with a large window, and rapid with a small window.

Fig. 8 has been used for about a year and a half in the steam power plant at Västeras, Sweden, belonging to the Royal Board of Water Falls. (N. Forssblad in *Combustion*, vol. 15, no. 1, July, 1926, pp. 33–35, 2 figs., *p*. A table is referred to in the article, but not given.)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

CORRECTION. The article entitled Experiences with Modern Stations which was abstracted in the July, 1926, issue of MECHANICAL ENGINEERING from *The Electrical World* of May 8, 1926, was not a report of a Committee of the National Electric Light Association but an original article appearing in the last-mentioned journal.

Engineering and Industrial Standardization

Pioneers in Standardization

THE death of Past-President Oberlin Smith on July 19, 1926, calls to mind the organization and work of The American Society of Mechanical Engineers' first "Standards Committee."

In 1889, twenty-nine years before the organization of the American Engineering Standards Committee, Mr. James W. See of Hamilton, Ohio, read a paper[1] before the A.S.M.E. in which he called this Society's attention in detail to the great need for the registration of standards in much the same way that inventions are registered. He proposed that Congress set up a "Bureau of Standards" for this purpose, and urged the Society to take steps to assist in bringing about such government action.

He was far sighted enough to point out that the following three classes of articles should be considered for such standardization: (1) articles which intermember, (2) articles designated by grade, and (3) rules of action (procedure).

In the long list which he gave in this paper we find such products as the following which have only recently received the status of American or Tentative American Standards. In fact some of them are still in the hands of the Sectional Committees organized to unify present practice and recommend a national standard. He urged the registration of standard dimensions for fire-hose couplings, circular saws and their arbors, screw threads for bolts and machine screws, bolt heads and nuts, machine-tool posts, bolting slots (T-slots), washers, rivets, wire rope, wire gages (26 in United States at that time), paper sizes. Under Mr. See's scheme "Any respectable representative body of any craft, trade, vocation, or business, having by due deliberative proceedings, adopted a standard, and having duly appointed a committee to attend to the government filing of said standards, may, upon due proceedings, file such standard in the Patent Office, and thereupon such standard shall be known as the United States Standard." He proposed further that a "Symbol should be legalized to this end. This device upon an article, followed by the standard number, would be equivalent to saying 'This car axle has its bearings in conformity to U. S. Standard, No. 728.' "

Mr. See was of the opinion that "Filing with the Patent Office, might include a specification, a drawing and sometimes a sample. Patent Office might criticize and call for changes in the proposed standard, to improve its quality or completeness. Notification through the Patent Office Gazette would permit those interested to enter protest or point out defects. Discussions might be submitted here as in Interference Proceedings. Other standards might follow. There would be no abrogation of a standard once filed, the new standard taking its own new number."

During the discussion which followed the presentation of this paper Mr. Oberlin Smith, former Manager of the A.S.M.E., agreed in the main with the proposal but was unwilling to allow an individual or an association of individuals to form a standard on its merit. He preferred a large and well paid national commission which would consider proposed standards.

In closing the discussion Mr. See expressed the belief that the government should not do the standardization work but simply make and hold the record. He would have the changes in the future made precisely as the standard was originally made but because of the registration they would be made with more understanding and would be based on a common experience. He stressed not the lack of standards but the lack of record of standards and then proposed that a committee to investigate this subject be appointed by the President. This resolution was passed and the following three members of the Society were appointed to constitute this committee: James W. See, Hamilton, Ohio; Coleman Sellers, Philadelphia, Pa.; Oberlin Smith, Bridgeton, N. J.

[1] See paper entitled Standards, prepared by James W. See of Hamilton, Ohio, presented to The American Society of Mechanical Engineers in 1889 and printed in A.S.M.E. *Transactions*, vol. 10, 1889.

This special committee made its first report at the November, 1889, meeting held in New York. It reads as follows:

TO THE PRESIDENT:

We, your committee appointed at the Erie meeting to take under consideration the subject of a Governmental Bureau for the record of Standards, beg leave to report as follows:

First. We give our earnest recommendation to the project, and advise that it is a subject to which the Society can with propriety lend its utmost aid and encouragement;

Second. We present herewith a resolution looking to the appointment of a committee to further the matter of Congressional legislation on the subject, and looking also to practical operations under the legislation in case the same should be secured, and we recommend the passage of the resolution;

Third. We present herewith a resolution giving the Society's approval of the installation of such a Governmental Bureau, the object of this latter resolution being to provide the above committee with the Society's endorsement of the project, and we recommend the passage of the resolution;

Fourth. We present herewith the draft of an Act of Congress looking to the establishment of the desired bureau, and suggest substantially such an Act as the basis of the operations of such committee.

Respectfully submitted,
JAMES W. SEE
COLEMAN SELLERS
OBERLIN SMITH

The Committee's first series of resolutions was then presented for adoption by the Society, and was carried in the form below:

RESOLUTIONS

RESOLVED, by The American Society of Mechanical Engineers:

First. That a committee of three be appointed by the President, such committee to be known as the Committee of Standards;

Second. That when, from any reason, vacancies arise in such Committee, the President shall, on notice thereof, fill such vacancy by appointment under the Rules;

Third. That it shall be the duty of such Committee on Standards to use all reasonable efforts to secure such Congressional legislation as will provide a Governmental Bureau of record wherein may be entered respectably recognized and approved standards, for the promotion of uniformity in the products of arts, in technical customs, and in nomenclature;

Fourth. That in the event of such legislation being secured and such bureau being provided for, it shall become the duty of such Committee on Standards to file, on behalf of this Society, applications for the entry of such Standards as the Society may hereafter from time to time approve for record;

Fifth. That properly certified bills for postage, stationery, and other expenses incurred by such Committee of Standards in its efforts to procure the Congressional legislation herein looked to, be paid out of the funds of the Society in amounts limited at the discretion of the Council.

The second series of resolutions, which was thought to commit the Society to an opinion or policy, and which would be transmitted as introducing the proposed bill to Congress, was as follows, after suggested amendments were made by the meeting and accepted by the Committee:

RESOLUTIONS

RESOLVED, by The American Society of Mechanical Engineers, having more than one thousand members engaged in the manufacturing industries of the country and in allied professional pursuits:

First. The time has arrived when practical Standards of uniformity based on a common understanding are essential to industrial business and professional pursuits;

Second. A large number of practical Standards of respectable recognition and approval are now in use, but without authentic record;

Third. Systematic procedures in the matter of recording such approved Standards, would tend toward the inauguration of many other greatly needed Standards, tending toward uniformity and interchangeability of merchantable products, in improved codes and signals, and in scientific nomenclature;

Fourth. Provision should be made, under Governmental auspices, for a place of record for Standards having respectable recognition or approval regarding their fitness for such record.

Later in November, 1921, the Chairman made the following progress report:

TO THE PRESIDENT:

A bill to provide for the registration of standards, in accordance with the resolution passed by the Society, was presented to the lower house of Con-

gress early in the last session, and was referred to the Committee on Patents, Butterworth, Chairman. Your committee appeared before that committee, and argued in favor of the bill, and the general sentiment, as unofficially gathered, was favorable. Later correspondence with Mr. Butterworth failed to get any responses, and the introducer of the bill, Mr. Morey, was appealed to, and reported that he had consulted with Mr. Butterworth who stated that he would report in favor of the bill. But he did not report, so far as your committee has been able to ascertain, and it is believed the omission was due to outside duties assumed by Mr. Butterworth. The session expired and the bill expired with it.

Your committee therefore asks for further time in which to try it again at the coming session of Congress.

The Society has before had experience in matters of this kind, and with the same result, it being found extremely difficult to arouse in Congress an interest in a matter which has no politics in it.

Respectfully submitted,

Hamilton, Ohio, November 2, 1891. J. W. SEE, *Chairman.*

The final report of this first A.S.M.E. Standards Committee was presented at the November, 1892, meeting in New York and reads as follows:

TO THE PRESIDENT:

Your Committee on Standards would report that the Bill regarding Standards matter came up in Congress; was referred to Committee on Patents; was argued before the Committee; was looked favorably on by the Committee as far as their expressions went; and the Chairman stated he would report favorably. The report was not made, and the Bill died at the expiration of the Congress. From an investigation of Congressional methods, your Committee is inclined to believe that measures of this kind will not receive satisfactory action by Congress until it has been before them for many years, and hardly then in the case of a measure possessing no more popular interest than the measure in question. Under these circumstances it is believed that the desired results can be more promptly accomplished through the medium of some special association, disconnected from the Government and disconnected from other societies; Governmental adoption of the work following later, when its utility and importance may have been established by the association which inaugurated it. Under these circumstances it is suggested that the Committee be discharged and the matter dropped so far as the Society and Congressional action is concerned. This report is made by the Chairman of the Committee on Standards, without consultation with his colleagues.

Respectfully submitted,
JAMES W. SEE,
Chairman 2d Committee Work.

Principles Recommended to Guide National Industrial Standardization[1]

UNIFORM GUIDANCE AS A CONDITION OF UNIFORMITY, OR, ALL THE NATIONAL STANDARDIZATION "UNDER ONE ROOF"

IT IS desirable that in every country there be only one national standardizing body, recognized nationally and internationally. In other words, there should be in every country, only one authority entitled to promulgate national industrial standards and one central office to guide matters concerning national industrial standardization.

STANDARDIZATION A DIRECT CONCERN OF PRODUCERS AND CONSUMERS

It is important to realize that national standardization is being done by collaboration of producers and consumers and eventually of other parties concerned and not by the officials of the standardizing institution. Consequently it is advisable not to undertake any standardization unless desired by those concerned and to discontinue a standardization work when found that there is inadequate interest for same. That, of course, does not mean to imply that the ascertaining of such interest by investigation, propaganda, or other appropriate means be abstained from.

Though producers and consumers are the main groups of interests to be respected in the standardization work, it will often be found necessary or useful to invite collaboration of representatives of institutions and individuals of impartial character, such as scientists, expert advisers, educators and the like. In some cases collaboration of representatives of employees may be needed, as for instance in cases dealing with safety codes, working rules, and the like, where interests of employees are directly concerned or, in general, where the expert advice of workers is desirable.

In classifying interests in standardization into the main groups of producers and consumers, the fact should not be overlooked that in practice every producer is often in the position of a consumer and vice versa, and special attention should be given to this variable character of representatives in technical committees. This concerns especially representatives of commerce and commercial distributors.

It should be considered a matter of foremost importance that technical committees be composed of not merely experts but rather of possibly fully authorized and responsible delegates of firms, organizations, and institutions they are representing. Delegates of large and important concerns should therefore be requested to assist and, if necessary, be assisted in organizing meetings at home prior to the meetings of the national technical committee, this being especially desired in the case of collaboration with Governmental departments.

Though the officials of the standardizing institution, including the central office, are not standardizing, they are indispensable in the complicated tasks of bringing interested parties to a desired agreement.

THE PRINCIPLE OF DEMOCRACY

It is essential that every group of those interested be given a free and adequate opportunity to participate in the development of national standards, that technical committees be composed of representatives of all important groups of those interested and that all interests be possibly equally represented.

Greatest care must always be used when arriving at decisions in committees where, for some reason or another, some important group of those interested is not adequately represented, as it often happens in cases where either the producers or consumers are lacking organizations, as for instance, handicraft, public as the final consumer, etc.

THE ECONOMIES CONTEMPLATED ARE THE BEST GUIDE

Every standardization work should be guided, throughout the whole process of its development, by estimating again and again the economies, which it is desired to achieve. The standardizers must not fail to see and repeatedly estimate savings which the standardization promises. Where this is not being done, the danger is imminent that standardization may become unproductive and that the efforts of standardizers will be wasted. Further, national standardization cannot be carried on successfully unless standardizers succeed in convincing producers and consumers of its material and other benefits.

FIRST WORD TO PRODUCERS

As a rule it will be found the best practice to invite manufacturers to bring forward and develop standardization projects. Being able to judge the possibilities of manufacture and knowing at the same time the requirements of consumers they are fully qualified for suggesting sound standardization projects.

The above should in no way prejudice the importance of consumers and other interested parties in standardization work.

DO NOT ISSUE A NATIONAL STANDARD BEFORE SECURING ITS ACTUAL PREPONDERANCE

A national standard should not be issued unless there is adequate evidence that it will be used and required by an overwhelming majority of industry and consumers, this being especially important in cases where the alteration of the prevailing practice is contemplated.

A safe opinion, that a proposed standard is supported by an actual majority, that is to say, that there is a majority sufficient to secure its future preponderance in practice, can be arrived at, rather by unbiased investigation than by voting of members of a committee.

It is undoubtedly a safer and better way to secure an actual majority in favor of a standard before issuing it, instead of relying upon any future propaganda and publishing a standard of uncertain value.

It would certainly be a failure, if a new standard should remain in a minority, thus complicating instead of simplifying matters.

[1] This memorandum was circulated by Mr. B. Rosenbaum, Secretary of the Czechoslovakia Standardizing Associations, to the national standardizing bodies. Copies of it were in turn distributed to the members of the Main Committee of the American Engineering Standards Committee on July 19, 1926.

A VOLUNTARY AGREEMENT BETTER THAN COMPROMISE

As, in general, in every technical committee the producers and consumers will be found in opposition, it may be considered as a safe measure of a subject being sufficiently treated, when both of the parties named cease to consider the worked-out project a compromise and begin to regard it as the best possible solution arrived at under the prevailing circumstances.

Though the practice described above may require much time and patience, it will be found one of the best and safest means of assuring the future life of a standard. As a matter of fact, standards developed in this way will live already before being promoted and published. They will hardly need any further propaganda for introducing them into practice, a fact, which alone may justify the somewhat greater efforts spent on their development.

KEEP NATIONAL STANDARDIZATION FINANCIALLY INDEPENDENT

Industry should, in the first place, be called upon to finance national standardization. In doing so, industry may safely regard it as an industrial undertaking managed on a commercial basis, which, in the long run is rather necessary, if any standardization *ex cathedra* be avoided and the standardizing office be safeguarded from doing unproductive work.

However, the above does not imply that the Government and others, to whom national standardization is rendering services, should not pay their share.

NOT TO MAKE A BUSINESS OF THE SALE OF STANDARD PUBLICATIONS

It should not be encouraged to rely upon the means derived from the sale of standard publications for the financing of the central office, as this might easily tempt to detrimental haste and superficiality in standardization work.

RELIABILITY OF NATIONAL STANDARDS AN INTERNATIONAL REQUIREMENT

From the point of view of international standardization it is most desirable that the national standards of individual countries can be relied upon as representing the majority of practice in those countries. It will enable such standards to be more readily accepted by other countries. It will by the mutual influence of one country upon another, help to elevate confidence in the national standardization work.

The very fact that the building of international standardization upon the organization of national standardizing bodies has been generally accepted, makes it evident that the responsibility for national standards should be extended internationally.

Standardization of Trimmed Paper Sizes Proposed

The United States Independent Telephone System, a member of the American Engineering Standards Committee, has formally requested the latter to authorize the organization of a sectional committee whose duty it would be to develop a coördinated system of trimmed paper sizes. It further suggests that the scope might be broadened to include a coördinated system of punchings for standard sizes of paper to fit standard binders and files.

In its letter to the A.E.S.C. the United States Independent Telephone System points out that the situation in this country with respect to the matter of coördination of trimmed paper sizes is very unsatisfactory and there is much need of the development of a system of coördinated trimmed paper sizes.

This great need has been made especially clear as a result of the study of the A.E.S.C. Special Committee on Size of Publications for Standards and the great difficulty of agreeing on any definite sizes on account of the enormous number of uncoördinated sizes in use.

International Standards Association

On September 21, 1926, in London, the official delegates of seven countries will gather to ratify the formation of an international standards association and to put the finishing touches on the constitution which was formulated during the Third International Standardization Conference which was held in New York, April 14 to 22, 1926.

The United States will be represented at this meeting of the Provisional Committee by Mr. C. E. Skinner, Chairman of the American Engineering Standards Committee.

Correspondence

CONTRIBUTIONS to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

Some Remarks on Engineering Standardization

TO THE EDITOR:

Dr. Mathews' address on Present Tendencies in Engineering Materials[1] brings up a vital question as to the real interpretation of the term "Standardization." Undoubtedly specifications are often exaggerated with unnecessary detail, while real progress in design and in the improvement of materials is curtailed by tendencies toward "overstandardization." On the other hand, a little reflection will, the writer thinks, convince most of us that the advancement of real and effective standardization goes hand in hand with engineering progress.

The importance of standardization is of course entirely relative, it being dependent upon the particular line of product, its method of manufacture, etc. Standardization is in a sense the engineering phase of production, and its importance is practically proportional to the relative importance of the production aspect in the manufacture of a given line of goods. On the other hand, standardization is often equally important when viewed from the operating and maintenance side. Thus there are powerful forces operative in the further promotion of engineering standardization, such as the linking up of standardization with production, resulting in the lowering of the cost of manufacture. In addition, further standardization can be applied in particular phases to the aggregate of industry, resulting in interchangeability of goods and preventing needless repetitions and continuous arguments while paving the way for a closer coördination of the engineering design and manufacture of a variety of goods, reacting mutually to the benefit of each separate element.

The engineering phase of standardization, the writer believes, offers one of the most unique technical applications of analysis and general science to engineering. In fact, to a considerable extent it seems that the true difference between science and technical engineering lies in the latter's linking of applied science to the standardization aspect. In the average manufacturing plant the very importance of an engineer hinges in the final analysis, on his relative weight in production, such as in improving manufacturing methods or in boiling down engineering design to a production basis, and at the same time effecting best possible compromise of technical analysis and scientific progress with the present status of the art.

It is a mistaken idea that standardization discourages engineering originality and slows up scientific progress. Though standardization may not actually follow immediate scientific progress, in a continuous curve, it does, nevertheless, move in steps, and the length of a step before a new change takes place is

[1] Mechanical Engineering, August, 1925, p. 791.

governed by economic considerations of scientific development outweighing the value of a given present method of production. Engineering in manufacture and design is based on a compromise, and this compromise offers fully as great a chance for originality as does the development of science itself. It continually coordinates with the latter as well. Its other aspects are similar to what we admire in ordinary life in the form of a well-balanced mind.

Strange to say, the broad aspect of standardization is a major research problem. It offers one of the greatest fields for a research department in the average manufacturing corporation. If one phase of research is devoted to the subject of standardization, it comes under immediate observation from a dollars' and cents' aspect, and in these "days" this point of view cannot be overlooked.

The improvement of the product, and the quality of material, etc., in no way conflicts with standardization. Real standardization should go hand in hand with real scientific progress, but it does offer a weighing device and constraint for the proper estimation of the relative importance of a given progress in science, and brings about a balance between economic and scientific advancement.

R. EKSERGIAN.[2]

Philadelphia, Pa.

The Engineer and the English Language

TO THE EDITOR:

One thought relating to the improper use of English by engineers which has not come out in the two recent editorial comments in MECHANICAL ENGINEERING seems to me to have a vital bearing on the subject. This is: the language of engineers—or anyone else for that matter—must be such as can be grasped by those to whom they speak.

Young engineers as a rule enter industry in some capacity that brings them in close touch with relatively uneducated men. To be understood, and to keep from being termed "high brow," the young engineer must adopt some, if not all, of the current idioms and slang. Also until recently business correspondence has been couched in rather stilted and prosaic forms. It was considered poor taste to try to make a letter original in composition.

Here, then, are two factors that tend to spoil the good work of English instructors. Their effect is soon seen in the speech of the young engineer, and frequently persists throughout his life. In time he may devote a little effort to making his written reports, papers, and the like rhetorically correct, but the writer believes the effort would be more effective were it devoted to perfection of speech, for it is through this medium that most projects must be initially presented; and it is by a man's speech and actions that the all-important "first impression" is given.

Therefore, while the writer agrees with the general proposition that any cultural reading will help, he would offer this additional suggestion: that those who bemoan the fact that their English is not all it should be, remember that there are enough words of exact definition in the language to express almost any idea. And further, that in their daily speech they endeavor to use these words rather than some of the vague slang that flows so readily, and which requires so much explanation to clarify a meaning. The use of a proper word tends to conciseness, which is what slang is supposed to do, but seldom succeeds in doing.

JAMES ELLIS.[1]

Kingsport, Tenn.

[2] Engineer, Baldwin Locomotive Works, Mem. A.S.M.E.

Contents of Spherically Dished Heads of Storage Tanks for Various Depths of the Liquid and Various Angles of Tilt of the Tank

TO THE EDITOR:

Storage tanks are tilted to insure perfect drainage. Acting upon a suggestion of Dr. O. R. Sweeney, head of the Chemistry Department, Iowa State College, the writer undertook the computation of tables to determine the liquid contents, expressed in R^3, of heads A and B, Fig. 1, for various values of d and d' for various angles of tilt α, of storage tanks with spherically dished heads.

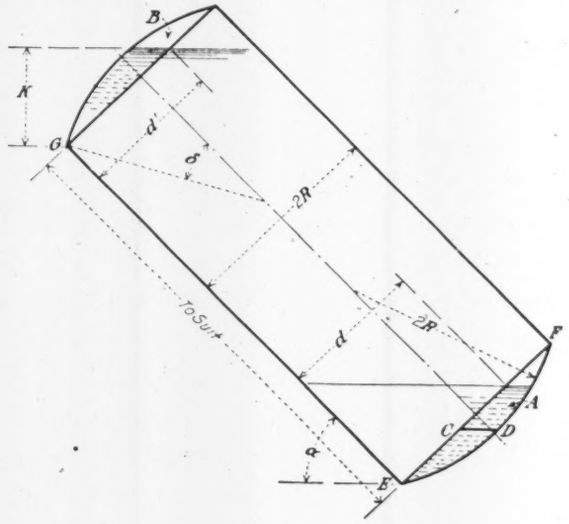

FIG. 1 SKETCH SHOWING PROPORTIONS OF SPHERICAL HEADS FOR WHICH TABLE 1 IS COMPUTED

TABLE 1 CONTENTS OF SPHERICAL HEADS IN TEN-THOUSANDTHS OF THE CUBE OF THE TANK RADIUS

(Radius of head = 2 × radius of tank. Heads A and B as in Fig. 1)

d/R and d'/R	Angle of Tilt α in Degrees: 5		10		15		20		25		30		35		40		45		50		55		60	
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
0.0	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
0.1	15	13	16	13	17	12	18	12	19	11	21	11	23	11	25	10	29	9	35	9	45	8	61	7
0.2	71	66	75	63	79	61	83	59	89	56	95	54	103	52	112	49	126	46	145	43	175	40	222	37
0.3	184	170	191	164	200	159	210	153	221	148	235	142	250	135	270	129	295	122	330	115	380	107	454	99
0.4	361	327	364	317	379	307	395	297	413	287	435	276	459	265	489	253	526	240	576	227	643	212	737	196
0.5	581	536	589	523	611	506	634	490	658	474	688	458	720	440	759	421	807	402	868	381	949	357	1055	331
0.6	846	791	861	776	888	750	917	728	948	707	984	686	1023	660	1069	634	1125	607	1194	576	1282	542	1396	504
0.7	1150	1085	1169	1065	1202	1035	1235	1008	1270	981	1309	954	1356	921	1406	888	1467	851	1541	811	1632	766	1747	714
0.8	1482	1410	1506	1386	1542	1352	1578	1320	1616	1288	1656	1255	1706	1217	1760	1177	1823	1132	1897	1082	1987	1025	2098	960
0.9	1834	1757	1860	1731	1898	1694	1936	1659	1975	1623	2012	1584	2065	1542	2119	1495	2181	1443	2253	1384	2338	1317	2441	1237
1.0	2195	2116	2221	2089	2260	2050	2297	2013	2335	1975	2376	1934	2423	1887	2474	1836	2532	1778	2598	1712	2676	1634	2768	1542
1.1	2553	2477	2579	2450	2616	2412	2652	2374	2688	2335	2726	2298	2768	2245	2815	2191	2867	2129	2926	2057	2993	1972	3073	1869
1.2	2900	2828	2924	2804	2958	2768	2990	2732	3022	2694	3055	2654	3093	2604	3133	2550	3178	2487	3228	2413	3285	2323	3350	2212
1.3	3226	3161	3245	3141	3275	3108	3302	3075	3330	3040	3356	3000	3389	2954	3422	2904	3459	2843	3499	2769	3544	2678	3596	2563
1.4	3520	3464	3534	3450	3560	3422	3582	3393	3603	3363	3624	3326	3650	3287	3676	3241	3703	3185	3734	3116	3768	3028	3806	2914
1.5	3774	3730	3787	3721	3804	3699	3820	3676	3836	3652	3851	3622	3870	3590	3889	3551	3908	3503	3929	3442	3953	3361	3979	3255
1.6	3983	3949	3994	3946	4003	3931	4013	3915	4024	3897	4034	3875	4045	3851	4057	3821	4070	3784	4083	3734	4098	3667	4114	3573
1.7	4141	4127	4146	4119	4151	4109	4157	4100	4163	4089	4168	4075	4175	4060	4181	4040	4188	4015	4195	3980	4203	3930	4211	3856
1.8	4245	4239	4247	4235	4249	4231	4251	4227	4254	4222	4256	4215	4258	4207	4261	4198	4264	4184	4267	4165	4270	4135	4273	4088
1.9	4297	4296	4297	4295	4298	4294	4298	4292	4299	4291	4299	4289	4299	4287	4300	4285	4301	4281	4301	4275	4302	4265	4303	4249
2.0	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310	4310

The solution of this problem leads to an unwieldy formula. Mrs.

[1] Mechanical Draftsman, Tennessee Eastman Corporation. Jun. A.S.M.E.

McKelvey of the Mathematics Department, Iowa State College, called the writer's attention to the following definite integral which gives the volume of liquid in the head B for various values of K, the vertical depth of the liquid:

$$V = \int_{R \csc \delta \sin(\alpha - \delta)}^{K + R \csc \delta \sin(\alpha - \delta)} (R^2 \csc^2 \delta - z^2) \left[\cos^{-1} \left(\frac{R \cot \delta - z \sin \alpha}{\cos \alpha \sqrt{R^2 \csc^2 \delta - z^2}} \right) - \frac{(R \cot \delta - z \sin \alpha) \sqrt{R^2 - (R \csc \delta \sin \alpha - z \sec \alpha)}}{R^2 \csc^2 \delta - z^2} \right] \delta z$$

However, this integral may probably be more simply expressed in the form of a series.

Table 1 was computed by means of Simpson's first rule, the simplest of approximate methods. A tank of proportions as shown in Fig. 1 was chosen, and for a certain angle α the surface areas of the liquid for 19 equidistant stations between E and F (station 1 is at E, station 21 is at F) as for instance at CD, were computed. By means of Simpson's 1, 4, 1 rule, volumes between stations 1–3, 3–5, 5–7, ... were obtained, and Simpson's 5, 8, 1 rule furnished the volumes from 1–2, 3–4, 5–6, ... thus giving the 20 increments of volume wanted.

It was remarkable to note the fact that the total of the 20 separate volumes thus obtained only differed on the average from the known volume by 0.04 per cent. The error in each separate case was of about that same magnitude. The nature of these differences shows that they were not due to the methods used but to personal and computational factors. Partial volumes for head B were obtained directly from head A.

It may seem that the mathematical solution given above is more desirable on account of its being applicable to tanks of various proportions. However, the outlined method using Simpson's rules is just as general. The substitution of certain constants dependent on the tank proportions is carried out in the elementary formula for the surface areas of the liquid, and the rest of the problem remains the same.

For complete information on the application of Simpson's rule in the determination of areas, volumes, centers of gravity, moments of inertia, etc., almost any book on naval architecture will be found satisfactory. A point that has to be stressed in computations of this kind is that of systematizing the tabulation of volumes obtained during the work. There are so many similar values that tabulation is a great help. A Monroe calculator also proved of value.

DIRK DEKKER.[1]

Ames, Iowa.

Autogenously and Electrically Welded Boilers[1] and Containers

W. L. Warner.[2] When the statement is made that one process of welding is most efficient of any welding process of the present day, there is surely some question with regard to just what is meant by the word "Efficient." I note that in the second paragraph of this paper, the statement is made that the water-gas process of welding is the most efficient process of today. I am at a loss to understand just what this statement means. I would take it to mean that any thickness of plate could be welded by this process and that with some thicknesses of plate, it is impossible to weld by any other process. This seems rather strange and I would like to have an explanation of why this is so. This situation does not seem to prevail in this country.

With reference to the tests of welded flanges, the apparatus used to determine the strength of the specimen is indeed very ingenious. However, it does not appear to me that types 1 and 20 as shown in Figs. 1 to 6 are very practicable. It would seem to me that types 2 and 3 were more practicable for ordinary work and I cannot find any reason why type No. 1 should be stronger and better than type No. 2, or why type No. 20 should be any stronger than type No. 3. It would seem to me that there must be some difference in the welding operation which would in some way account for the test results. A discussion of the fracture of the test specimens and the appearance of the fracture would be probably very interesting in explaining the results.

With reference to the welding of reinforcing rings, the test results shown on page 605 appear logical with the exception of result which shows that type No. 4 is stronger than type No. 5. It would seem to me that if the same cross-section of weld were applied on type No. 5 that it would be stronger than type No. 4 but just how much stronger, I do not know. In general the results do not give any idea of the amount of weld, that is, with reference to the size of the bead, and undoubtedly this feature will determine in a great measure the relative strength of the different designs of joint, although I think that it is without question that type No. 23 is the strongest of all.

The entire discussion of the various types of strap weld illustrated in Figs. 35 to 53, inclusive, is very interesting, and I believe that the proposed investigation of the American Bureau of Welding and The American Society of Mechanical Engineers should include some of these features in their program of tests. However, there is one thing in connection with this report which strikes me very forcibly and that is the part that says so much with regard to electric arc welding as to where and when it should be used, and yet the number of test samples which are shown in the various tables is very much smaller than the number of autogenous-welded samples. For this reason, I do not see how anyone could definitely state that electric arc welding is better for some things than gas welding, especially since in many cases only one electric-welded specimen was tested as compared with a dozen or more gas-welded specimens. It appears also that tensile, bending, and impact tests which were made in 1923 and which are listed on the first page of this report state that a great number of the samples were welded using the Quasi-arc electrodes, and yet when we come to consider the welding of flanges and reinforcing rings and the other features of this investigation, we do not find very many electric-welded samples. We find also that the examples of welded boilers which are given in the latter part of this paper are, in the majority, electric welded, and only two of the seven illustrations given have any gas welding done on them.

It appears to me, therefore, that this investigation while excellent in the diversity of experiments and ideas tested, is rather unbalanced in that it includes a majority of gas-welded specimens and only a few electric-welded specimens.

This appears to me to be one more important reason why the proposed investigation of the American Bureau of Welding should be very broad and thorough in its scope. In addition to this, all of the details of welding and preparation of the samples should be shown so that anyone reading the report can understand why certain conclusions were drawn and then if necessary, draw his own conclusions contrary to those of the committee if the test results should warrant this procedure.

H. L. Whittemore.[3] I agree with you that the conclusions are not justified by the data submitted and that the number of electric-welded specimens is insufficient. It may be, however, that many more specimens were tested than are reported in these articles. I am at a loss to know what is meant by the water-gas process of welding and why it should be so highly recommended.

It seems to me that in the tests of welded flanges, No. 1 should be much stronger than the other methods of welding the pipes to the shell. In No. 1 you have a straight tensile pull on the weld. The bending where the shell and pipe meet is taken by forged metal. In Nos. 2 and 3 the severe bending stresses come in the weld metal. Any deformation may result in cracks which will

[1] Discussion of a Paper by E. Hohn, published in MECHANICAL ENGINEERING, June, 1926, p. 603, and reprinted in the Journal of the American Welding Society, vol. 5, no. 6 (1926), which also included the discussion.

[2] Industrial Engineering Department, General Electric Co.

[1] Iowa State College.

[3] Bureau of Standards, Washington, D. C. Mem. A.S.M.E. Comment on preceding discussion of W. L. Warner.

decrease the ultimate strength. We know that when dished heads are welded to the shell there is an advantage in having a skirt on the head so that the weld is back several inches from the knuckle.

I agree with you, in the main, on the welding of reinforcing rings. The cross-sectional area of the bead, of course, should be given to allow the strength of the weld to be estimated.

I am glad to know you agree that the discussion of strap welds is interesting and that this German report should be carefully studied before our investigation is started. Personally, I do not believe that the reports I have seen of the German investigations justify us in adopting their conclusions. These reports do, however, suggest variables which should be studied in our investigation if we expect to obtain the best form of welded pressure vessel.

My purpose in using the German tests in my paper before the American Bureau of Welding was to call attention to the necessity for trying everything which was proposed and eliminating the unsuitable methods of fabricating tanks.

John J. Crowe.[4] There are several points I desire to call attention to in the paper by E. Hohn on Autogenously and Electrically Welded Boilers and Containers.

It is stated that "of the personnel employed in the test, some were beginners; today probably better results would be obtained." This statement of itself vitiates and casts a serious doubt on all of the results quoted. In this country only experienced welders are entrusted with the responsibility of welding pressure vessels and work done by beginners should not be used as a standard of comparison for various methods of welding, or for the setting up of standards to be used by engineers in the design of pressure vessels.

In Table 1 the average tensile strength of the 67 bars autogenously welded, is 2.81 tons per sq. cm., or 39,800 lb. per sq. in. This is an exceptionally low value for welded joints and would indicate that inexperienced welders had been used. The average tensile strength obtained on several hundred coupons welded by several embryo engineers in our laboratory, using low-carbon steel rod, was well over 50,000 lb. per sq. in. The welds were of the V-type (type 35), made from one side only and machined to size before testing, whereas the paper states that the figures quoted are for reinforced welds. When alloy-steel welding rod is used the results obtained are appreciably higher and in many instances are over 65,000 lb. per sq. in.

There is no basis for the statement that electrically fused iron adheres better than autogenously fused iron. In all cases where the break occurs through a weld properly made, the path of the fracture is in the added material, and not where the added material has been joined to the parent metal. This last only occurs in improperly made welds of either type—gas or electric. If the author (E. Hohn) has had autogenously fused metal separated from the parent metal, it is a clear indication that proper penetration has not been obtained.

Of the three variables studied, that is, tensile strength, bend (ductility), notch-bar impact (resistance to shock), the autogenous welds gave the better results in all but the tensile tests, but the author has evidently disregarded these facts in drawing his conclusions.

In Table 6 the mean forcing-off pressure of 60 rings autogenously welded is 50.3 tons as compared with a mean of 47.4 tons for four electrically welded rings, and again in Table 8 the autogenous method as judged by the results given, is greatly superior to the electric method for the purpose of welding plates perpendicular to each other.

It would appear from the paper that the autogenous method has been employed for welding a number of pressure vessels and that they have, in addition to remaining tight up to a pressure of 68 atmospheres, 1000 lb. per sq. in., successfully stood the test of time and service which is the real criterion by which any method must eventually be judged.

In view of the fact that the welds were made by inexperienced operators, the paper can hardly be considered as a contribution to the art of welding and fades into obscurity in the light of the most excellent work done along similar lines at the Bureau of Standards, under the auspices of the American Bureau of Welding.

[4] Engineer in Charge of Apparatus Research and Development, Air Reduction Sales Co.

S. W. Miller.[5] It is very unfortunate that the figures were not translated into English units, because as they stand, they mean nothing to the vast majority of those interested in the subject.

It should be remembered in reading this article that the word "autogenous" is used to refer to gas welding in counterdistinction to the word "electrical," which refers to metallic arc welding.

In the limited time available I can only point out a few serious matters. The statement that, "electrically fused iron adheres better than autogenously fused iron" is without basis in fact, as a molecular union is produced in both cases, and more cannot be done. There is no difficulty in gas welding mild-steel plate so that test pieces will always break outside the welds.

It is stated that in autogenous welding, welding both sides cannot be done very well. This is also contrary to the facts. Statement is made that double V joints can only be made in thick plates $^3/_4$ in. thick or thicker. This is another statement that is not borne out by common practice.

A very significant statement is made as follows: "Of the personnel employed in the tests, some were beginners, but today probably better results would be obtained."

Many of these tests were made in 1914, others in 1921, still others in 1923. It certainly seems futile to compare the results of tests made twelve years ago with present-day practice, and the use of beginners would certainly seem to have been a mistake, unless their results were segregated and not included in the averages. Such an inclusion destroys the value of any comparison and does not give a true idea of the possibilities of welding.

The use of straps welded across the seams is not considered in German practice a desirable thing. This is in accordance with the statement of Dr. Max Fuchsel, chairman of the Welding Committee of the German Engineers Society at Berlin. This statement was made at the last meeting of the American Welding Bureau by Dr. Fuchsel in commenting on this article. He also stated that it is the endeavor of the German welding engineers to produce a weld of such good quality that it is not necessary to use straps. He also said that in Germany one of the serious objections to the strap seam was the stresses which were undoubtedly set up due to the local heating.

I would also point out that there is a wide difference in design stresses in the welded pressure vessels shown. In the case of Fig. 66, the stress at working pressure is 8000 lb. per sq. in. In Fig. 64 the working fiber stress is about 7200 lb., Fig. 65 the working fiber stress is 6400 lb., while in Fig. 70 the working fiber stress is 9400 lb. per square inch. If these are typical examples of design, two things seem apparent:

1 There is no unanimity of opinion as to proper design stresses for strap construction.

2 That as far as the figures go, the fiber stresses are so low as to indicate lack of confidence on the part of the designers as to the quality of their welds and the validity of their designs, even with the theoretically greatly increased strength due to the straps. I would also point out that from a commercial point of view the application of straps is impractical because of the greatly increased cost, there being from two to four times as much welding in the strap seams shown as there is in a straight welded seam.

I agree with Dr. Whittemore very strongly that there is considerable ingenuity shown in the methods of test devised to determine the strength of various types of welded joints, and this to me is one of the important points in the paper. Another important thing is that it is shown clearly that lap joints in a tank should not be used in longitudinal seams except under low fiber stress. The danger of lap-riveted longitudinal seams has been known for years and their use is restricted in all codes to conditions where there cannot be any serious fiber stress. Their danger does not lie in the weakness of the weld, but, as shown in the paper clearly, in the very high stresses in the material next to the weld, which would be of the same order in a riveted joint.[6]

In Fig. 32 in the paper, the fiber stress at a short distance from the weld is shown as 21,000 lb. per sq. in. when the pressure is 213 lb. per sq. in., while the theoretical stress at this pressure, considering

[5] Consulting Engineer, Union Carbide and Carbon Research Laboratory, Long Island City, N. Y. Mem. A.S.M.E.

[6] This and succeeding paragraphs contributed by Mr. Miller after publication of his discussion in the Journal of the American Welding Society.

the shell as a true circle, is only 9650 lb. per sq. in. This should be more fully investigated, but it is evident that with thick plates at high pressures the stresses will be even greater than those recorded.

It is pointed out that straps welded on one side behave similarly to lap-welded joints. Some of these strap seams are patented in Germany, by the author of the paper.

Reinforcing rings applied to test pieces show a very great increase in strength over the plain bar with a hole in it, an increase of about 40 per cent being noted. Further tests should be made in this direction, especially with rings on both sides of the test piece. Theoretical investigations should also be made because the test piece is not a thoroughly fair comparison with an actual shell, which has a much larger area over which the stress is distributed, while with the test piece, the stress is concentrated within its boundaries.

A reference is made to the contour of the heads in a pressure vessel. This is also a matter that requires careful investigation because it has been found that under high test pressures there are heavy stresses in the knuckle.

It is stated that in gas welding the seams are generally only welded on one side because the seam and the adjacent plate suffer if repeatedly heated. It has been many times observed that the welding on the second side refines the grain of the weld and adjacent metal on the first side welded, and therefore, does not injure it but improves it. It is also stated that the seams are thickened on the outside as much as possible, referring of course to the single V. This is bad practice because of the eccentric load imposed, the fiber stress on the inside of the shell being much increased where the reinforcement is heavy. The tests at the Bureau of Standards on pressure vessels which are described in Bulletin No. 5 of the American Welding Society show distinctly that it is perfectly feasible to gas weld on both sides of a shell.

It seems quite clear that there is no danger in fusion welding boilers carrying as high as 180 lb. pressure, and there is no good reason why this work should not be done. It undoubtedly will be, carried out as soon as its advantages are thoroughly appreciated and procedure controls will be developed for the work just as they have in other cases. Finally, it appears that all of the pressure vessels illustrated have been welded with electrodes having wound covering, and that bare wire was not used.

A.S.M.E. Boiler Code Committee Work

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below are given interpretations of the Committee in Cases Nos. 526 to 529, inclusive, as formulated at the meeting of June 4, 1926, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

CASE NO. 526

Inquiry: Is it intended that the deduction of 100 sq. in. for a manhole in the head of an h.r.t. boiler, shall apply in the 1924 revised edition of the Code only to the portion of the head below the tubes as was the case in the 1918 edition, or is it applicable to the portion of the head above the tubes also? In the 1924 edition of the Code the sentence has been so divided that the statement of exemption does not appear to be limited to any particular portion of the head.

Reply: The reduction of staying allowed by the second sentence of Par. P-218 was intended to apply only under the conditions specified in the first sentence, namely, to the portion of the head below the tubes.

CASE NO. 527

Inquiry: Assuming that a 6-in. hole is permitted by Par. P-258 of the Code, in the shell or drum of a boiler without reinforcement, inquiry is made as to whether there will be any objection to the use of a 7-in. hole without reinforcement in shells or drums 2 in. and over in thickness, with a diameter of 48 in., provided the concentration of stress at the edges of the hole does not exceed that experienced with a 6-in. hole, namely, between 16,000 and 17,000 lb. per sq. in.? With the construction contemplated, the normal stress in the shell would be from 5000 to 6000 lb. per sq. in.

Reply: The restriction in the Code of 6 in. as the greatest opening in a shell or drum without reinforcement, was not intended to apply to shells of thickness of 2 in. or over, and for the case in question, it is the opinion of the Committee that a 7-in. opening is permissible without reinforcement, provided the construction otherwise meets all the requirements of the Code.

CASE NO. 528

Inquiry: Will it meet the requirements of Par. U-53 of the Code, that calls for a handhole in a compressed-air tank 16 in. in diameter or over, if 16-in. air tanks are fitted with 2-in. threaded openings to be closed with standard 2-in. pipe plugs and suitably located so that light bulbs can be inserted to facilitate inspection of all parts? Attention is called to the fact that 16-in. pneumatic reservoirs as used for street-car air-brake purposes are too small to accommodate the usual type of oval handhole openings.

Reply: It is the opinion of the Committee that if, under the conditions named, the vessel is fitted with plugged openings of 2-in. pipe size, so located as to admit electric-light bulbs and facilitate inspection of the entire interior, the intent of the requirement in Par. U-53 may be considered as fully met.

CASE NO. 529

Inquiry: Will a steam-actuated fusible-plug device for indication of low-water level in boilers, come under the rules for the fire-actuated fusible plugs given in the Code?

Reply: The requirements and recommendations in the Appendix of the Code for fusible plugs, if used, are for fire-actuated fusible plugs and do not refer to those of the steam-actuated type. There is nothing in the Code to prevent the use of steam-actuated fusible plugs.

Group Drive and Individually Motorized Drive

(Continued from page 892)

with respect to the lineshafting. It is not convenient usually to place them at an angle, even though the machines might thus much better utilize the space available. This is where the inividually motorized tool is at a great advantage. The tool can be placed as best suits the convenience of the production routine, and, if upon trial it does not prove satisfactory, it is a relatively simple matter to move it to a point where this is true, a feature which is not possible in the case of the group drive.

A factor entering frequently into consideration is the availability of galleries located along the side walls of various buildings which are employed for machine shops. It has happened in the past that sometimes these galleries have been so designed with a relatively small headroom so that it is not possible to install overhead lineshafting at all. In such a case, obviously, the group drive is out of the question and the individual drive must be substituted therefor. This last, however, probably is of secondary importance, as it is present-day practice to employ just so far as is possible buildings which are well lighted, and galleries of small headroom are apt to be rather hot during the warm summer days. Hence this construction is avoided in small cities or towns where manufacturing plants are located. However, in larger cities with space at a premium, it is not uncommon to have this condition of affairs exist.

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Progress in American Aeronautics

LESS THAN a year ago it was still a common plaint that American aviation, while fully abreast of European developments from a military point of view, was lagging sadly behind in the matter of commercial operation of aircraft. At the present time it may almost be said that the United States is leading the world in civil aeronautics. This fortunate transformation is directly traceable to two thoughtful legislative actions of Congress.

The first of these acts, dating from 1925, allowed the Postmaster General to place contracts for the carrying of mail by air with private companies. As a result of this act twelve contract air-mail routes are now in operation; Boston-New York; Chicago-St. Louis; Chicago-Dallas; Salt Lake City-Los Angeles; Pasco-Elko; Detroit-Cleveland; Detroit-Chicago; Seattle-Los Angeles; Chicago-Minneapolis; Atlanta-Miami; Cleveland-Pittsburgh; Cheyenne-Pueblo. There is a certain satisfaction in the very names of these companies which show how rapidly the territory of the United States is being covered by a nation-wide service.

These air-mail contractors are of a responsible character, with the Ford Motor Car Company itself as one outstanding example of financial reliability. It is true that the majority of them are working under a daily deficit running into considerable amounts. But the financial backers of these air lines are business men who are willing to pay for operating experience. The traffic managers moreover are educating the public rapidly to the value of this new service on lines covering now more than 9000 miles.

It was not sufficient however to give these companies contracts and to allow them a large proportion of the excess postage entailed by air mail. The problem arose immediately as to how best to aid them in the pioneer stages. In Europe, air traffic is regarded fundamentally as a reserve for the military forces of the various nations, and enormous subsidies have been granted to the air lines as a logical consequence of this conception.

In the United States subsidies have not been asked for by either public or aircraft operators, and a much wiser policy than that of "pauperization" by subsidy has now been initiated. In the wake of the Lampert investigation of the House of Representatives and of the wise conclusions of the President's Air Board, there followed the Air Commerce Act of 1926. The Act provides for the appointment of an assistant secretary in the Department of Commerce, whose duties shall be first to provide for air-traffic regulation and second to establish such navigational aids as may be necessary for safe flight by night and in adverse weather conditions.

As soon as the act was passed, Mr. Hoover, the Secretary of Commerce, ably assisted by Assistant Secretary J. Walter Drake, set to work to make the provisions of the act a reality. They have quietly done a great deal of work in this direction, and the new Assistant Secretary William P. MacCracken, Jr., on assuming office, found a tentative plan of action and a skeleton organization at his command.

The problem of air regulation is one of great importance. It is important that the public be safeguarded, that a reasonable amount of inspection be organized, that pilots engaged in the arduous duties of cross-country flying be well-qualified men, that ground engineers and mechanics be trained so as to give plane and engine the maximum of fitness. It is also important that the rules and organization to be administered by the new assistant secretary should not be of too exacting and bureaucratic a character. We may have the fullest confidence in Mr. MacCracken's ability to steer a wise middle course. An Army pilot during the war, a lawyer of great ability and reputation, Mr. MacCracken is familiar with the legislation presented to Congress in aviation matters, and is perhaps one of the best informed men on the subject in the country. At the same time he is fully aware of the needs of the new industry and may be trusted to set as his goal the wisest policies in matters of air regulation.

At the same time we may confidently expect energetic and positive action in the establishment of navigational aids. Broadly speaking navigational aids to aircraft may be divided into lighting, radio direction, and weather reporting. The new Commerce Air Service will have approximately $300,000 to spend in its first fiscal year for the provision of such aids. On those air contract lines which are to be flown at night advantage will be taken of air-mail experience to set up a system of giant beacons, spaced at ten-mile intervals, by the aid of which the pilot will be able to fly with real confidence in all but the worst fogs. Later the present lighting system of the air mail will be taken over. In the ensuing fiscal year, radio direction beacons will be set up, which will send signals to the aviator whenever he deviates from his course, and render flying safe even when the lights fail to penetrate through mist or fog. In coöperation with the Weather Bureau, the new Commerce Air Service will at an early date supply the flier with full weather reports, in particular warning him of approaching storms or conditions of low visibility. Two airways have already been designated as national airways by the President; others will follow. In a comparatively short time, the United States should be provided with a system of public airways, free to the use of all, laid down on the logical pathways of air travel, and as useful as the public roads.

The Prince Pays His Respects to Science

THOSE born to rule the British Empire must be all things in all fields. The Prince of Wales is the President of that august body of savants, the British Association for the Advancement of Science, and at the opening of its recent Oxford meeting on August 4, 1926, he gave an address which from many points of view is an interesting document. As an onlooker in scientific work, he spoke of his firm conviction "that if civilization is to go on, it can only progress along a road of which the foundations have been laid by scientific thought and research."

As an instance of contrast between modern understanding of the debt to science and that eighty years ago, the speaker recalled the objections in 1832 of an Oxford scholar because of the award of honorary degrees to a "hodge podge of philosophers." These men were David Brewster, Robert Brown, John Dalton, and Michael Faraday, each of whom has left a permanent impress in his own special branch of science. He pointed out that today two million workers of Great Britain are living on the brain of Faraday and to his discoveries many millions more owe the uses of electricity in lighting, traction, communication, and industrial power. The Prince then related some of the achievements which have been made by coöperation between the laboratory and the government. He emphasized the contributions of scientific research in agriculture and in medicine. He pointed out that scientific research properly applied and carried out is never wasted and

that the coöperative attitude of the state toward science once achieved, makes for the easing of the paths for the advancement of science in many directions. It marks a definite step in human progress taken after long hesitation but in itself new and, because it is new, we may believe for some reason that we live not merely in the age of science but at the beginning of it.

In 1924 at the Washington Meeting of the American Association for the Advancement of Science, President Coolidge and Secretary Hughes, made similar pronouncements. These public statements by nations' rulers are effective in calling to public attention the importance of science and they are welcomed by workers in science and engineering. The automobile, radio communication, and aerial navigation have brought home the realization of man's dependence on science for his well being and his luxuries. But there is much to be done and in this country the American Association for the Advancement of Science is bending its efforts to remove those disadvantages which impede scientific progress wherever they may be.

Water Committee Denies Approval of Products for Water Treatment

THE Boiler Feedwater Studies Committee has been formed for the purpose of studying the various processes in the purification of feedwater employed in steam stations and on railroads. Research work is being carried on to determine the fundamental principles of certain phenomena which take place in steam boilers. The results of the findings of the various committees are to be published from time to time in the journals of the associated engineering organizations affiliated in this work.

The Committee has not given approval to any systems of feedwater treatment and has not approved any product manufactured for internal treatment of boiler waters. Advertisements indicating this are untrue and misleading. The Boiler Feedwater Studies Committee is sponsored by the following organizations:

The American Society of Mechanical Engineers
American Water Works Association
American Railway Engineering Association
National Electric Light Association
American Society for Testing Materials.

(Signed) S. T. POWELL, *Chairman,*
Executive and Editing Committee.

Smoke Must Go

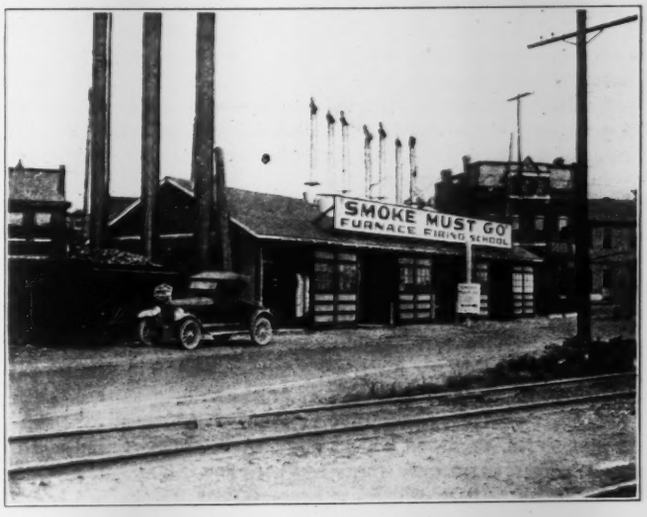
UNDER the leadership of the St. Louis Section of the A.S.M.E the citizens of St. Louis have entered upon a determined campaign to remove the smoke nuisance of that city. A large active organization has entered upon this task with the full understanding that successful smoke-abatement activities must be as unremitting as street cleaning or police work.

An engineering estimate of the cost of the smoke nuisance to St. Louis gave a total figure of $15,500,000 per year. With this figure before it and a keen appreciation of the desirability of a city's reputation for cleanliness it has been a simple matter to arouse enthusiasm in the far-reaching program which will require the expenditure of $60,000 a year.

This campaign is the outcome of a meeting held early in February under the auspices of the St. Louis Section, A S.M.E., which was addressed by O. P. Hood, Chief Mechanical Engineer of the Bureau of Mines, and Chairman of the Joint Committee on Smoke Abatement organized by the Fuels Division of the A.S.M.E. From his own experience in solving smoke-abatement problems Mr. Hood outlined the character of the work that must be done to cleanse the St. Louis atmosphere. The enthusiasm aroused by his address resulted later in the organization of the Citizens' Smoke Abatement League. This organization now has representatives of fifty civic, industrial, and engineering bodies in St. Louis. At a meeting in April officers were elected and the general plan of attack was adopted.

The work is divided into five sub-divisions devoted to survey, educational, technical, publicity and promotion, and finance. The chairman of each of these sub-divisions with the elected officers make up the executive committee.

The first steps to be taken by this organization include the conduct of a complete survey of the smoke-producing apparatus in the city. The technical division will attack the problem of burning Illinois coal without smoke and this division is divided into sub-divisions to deal with the various industrial problems of St. Louis. The educational division plans to "sell" smoke abatement to every individual in the community, carrying its message personally into every power plant, every heating plant, and every home in the city, not once but repeatedly, until the smoking chimney is looked upon as a disgrace to St. Louis and those responsible for it. The publicity and promotion division is authorized to keep the message of smoke abatement constantly before the citizens of the city. The finance division must provide the necessary funds.



EXTERIOR VIEW OF FURNACE-FIRING SCHOOL

INTERIOR VIEW OF FURNACE-FIRING SCHOOL

In accord with its policy of educating the general coal-using public the League established a furnace-firing school for the benefit of those who desire to learn how to fire a furnace properly. This school contains seven examples of furnaces and three of house heating boilers, so that experience with any type of apparatus can be secured. In the first six weeks of operation, attendance at the school was about 1200, each visitor spending from about one-half hour to an hour. The coal industry of St. Louis has sent one hundred men to secure experience on the firing of smokeless boilers. These men are being trained as demonstrators to assist the householders when the heating season opens. The interest of the flat and apartment janitors' union has been secured and the six hundred members of this organization started attending the furnace school during August. Ten nights were set aside for them and they came in groups of from fifty to sixty.

Basel Power Conference

IN JULY, the International Exhibition for Inland Navigation and Utilization of Hydraulic Power was opened at Basel, Switzerland, celebrating the opening of Basel's new port on the Rhine. The exhibition will remain open through September and one of its features will be the Sectional Meeting of the World Power Conference. An elaborate program has been prepared for this conference and thirty-two countries will be represented by the authors of over seventy papers, five of whom are from the United States.

At the close of the 1924 World Power Conference at Wembly, a second conference was suggested for 1930 and a permanent international executive committee was appointed to consider future plans. This committee received the invitation from Switzerland and granted permission for a sectional meeting at Basel. The holding of a plenary meeting was deemed unwise so shortly after the successful Wembly event.

The United States will be well represented at Basel and The American Society of Mechanical Engineers has appointed three members to represent it should occasion require.

Engineering Education

INCREASED interest attaches to the second series of reports issued by the Society for the Promotion of Engineering Education covering the work done by its committees under the grant of the Carnegie Corporation for an investigation of engineering education. The first series, reviewed a year ago,[1] were largely statistical, but those who read them were impressed with the belief that something tangible in the nature of definite conclusions and recommendations would shortly be forthcoming. The present series of reports, presented at the annual meeting of the society, Iowa City, June 16 to 18, has justified this belief.

W. E. Wickenden, Director of Investigation, in his report to the committee and to the society, summarizes the position taken as to the scope and function of the engineering college as follows:

1 The engineering college is not the exclusive instrumentality of engineering education. In a satisfactory scheme it would share the field with schools giving a more intensive form of technical training, and would be supplemented by means for extending professional training beyond college. The present unbalanced situation is a source of inefficiency and confusion.

2 The undergraduate status of the engineering college seems to be sound in principle and practice. Few institutions will give over to other colleges the entire pre-professional portion of their curriculum. It is desirable that there should be a closer articulation between the programs of engineering colleges, junior colleges and arts colleges.

3 Engineering colleges do not claim jurisdiction over all realms of higher technical education, but chiefly those relating to public works and industry. They assert their claim to a primary interest in the field of industrial operation and management. They undertake to protect the name engineering from indiscriminate use.

4 An engineering college cannot offer a complete professional discipline. There should be a sharing of responsibility for shaping the course of engineering education with professional and industrial bodies, and a working collaboration for closing the gaps in the present educational scheme.

Turning to internal problems, the report deals with student and teaching personnel. It points to the necessity of more effectual educational guidance and publicity, problems which reach back into the home and secondary school, and commends a selective standard of admission, "believing that the proper intellectual demands of engineering education are more severe and its demands on occupational aptitude are more definite than those which may reasonably be applied to collegiate education in general." For the student himself the report recognizes the severity of the adjustments to the new mode of life which faces him, and the responsibility which the college has in the handling of a situation which is further complicated by the problem of failures and eliminations, many of which might never occur with proper adjustments of admissions, curricula, and unbalanced facilities, and from which much might be saved through wise and sympathetic counsel. A similar problem, no less severe and resulting in maladjustments, presents itself when the graduate becomes the employee. The report points out that a saner placement system can be hoped for only through organized effort on the parts of college, industry, and engineers. With regard to the teaching personnel the following paragraphs are quoted from the report:

[1] See MECHANICAL ENGINEERING, August, 1925, pp. 687–688.

When a German is invited to a professorship, he is chosen to be primarily one of the principal agents for the advancement of his division of engineering, science and art, and incidentally a teacher of youth. All illusions aside, do we not ask a teacher of engineering to be first a pedagog, second an auxiliary officer of administration and discipline, third his own combination secretary, file clerk and technician, and fourth—when there is any fourth—something of a creative factor in the realm of knowledge he professes. Granting that such a career is both honorable and useful, why should we expect it to appeal to engineers of distinguished ability?

Here again we must appeal for moral and material support to the engineering profession and industries at large. We must create greater concern in these circles for the caliber of our teaching personnel and for the use of the latent creative possibilities in these colleges. Backing from these sources may go far in bringing administrative heads to recognize the inherent differences between engineering professorships and academic professorships in general and to provide financial rewards and professional opportunities accordingly.

Under Curriculum, the major problem of the investigation, it is gratifying to learn that "the verdict of all parties consulted, teachers, graduates, engineers, and industrialists is definitely against specialized training for undergraduates." In commenting on functional vs. technical plans of instruction, the report states:

It would seem that the functional scheme has distinct merits and deserves recognition in the curriculum plan, but as a secondary rather than a primary feature. We are inclined to favor preserving the framework of the present major technical curricula, and creating a bi-lateral division on functional lines in those fields which are associated with large-scale industry, particularly electrical and mechanical engineering. One side of the program might provide the more concentrated technical training suitable for the research, design, planning and advisory aspects of engineering, and the other a program of less technical concentration and suitable for the constructing, operating, and commercial aspects. In general, we do not favor a distinct curriculum to give special stress to the economic and managerial aspects of engineering, which we regard as common to all the major technical divisions.

Speaking of the study of economics, the report states:

In an ideal conception of the curriculum, economics would be ranked with the physical sciences as one of the foundations of engineering. The present formal treatment of the subject does not seem to be based on such a conception, but rather that of a collateral subject of primarily cultural values. As long as this conception prevails, the provision of time for work in formal economics seems adequate, in most cases. If the treatment of the subject could be made to articulate more closely with the interests and problems of engineering, a more generous time allotment would be warranted.

Greater emphasis on the economic aspects of engineering subjects proper seems clearly to be feasible and desirable. The need seems to be for only limited modifications of the factual materials in these courses, dealing specifically with costs and other economic details, but a closer association of economic and physical elements in engineering problems and in the criteria for their solution.

The work of the Board of Investigation and Coördination has brought many facts to light which call for future action. In looking toward the future, Mr. Wickenden concludes his report as follows:

This report has given repeated stress to the problem of engineering colleges which call for concerted action. When recapitulated, they make an imposing array. There is the problem of a better balanced system of technical education; of articulating curricula in engineering with those of other colleges; of defining an acceptable function and scope for engineering colleges; of carrying out an adequate program of educational guidance and publicity; of a selective basis of admission; of mutual coöperation with secondary schools; of providing constructively for the personal and occupational adjustments of students, and especially for a more satisfactory placement procedure; of strengthening the status of engineering teachers; of reasonable differentiation of curricula in accordance with technical and functional activities; of relief from over-crowded programs; of provision for more effective post-scholastic training and of more adequate but duly safeguarded graduate work; of the terms of award of professional degrees; of giving greater flexibility and selectivity to our program; of introducing a more distinctive professional tone to the upper years of the work; and of more adequate emphasis on the economic basis and content of engineering. I can assure you that it is limitations of time and not of material that prevents this list from being largely extended.

The conclusion seems fairly inescapable that we have already gone about as far in developing a national system of engineering education as the initiative of individuals and of isolated institutions can be expected to carry us. The present investigation has revealed a strong mood for concerted action among the colleges, provided their final autonomy is fully safeguarded. What is desired is not standardization but concerted initiative. It seems equally plain that we are faced by many problems that the colleges alone, however effectively united, can not solve, and that the colleges, the engineering societies and the engineering industries are being drawn together by the recognition of mutual problems and interests. What is wanted is effective collaboration, and not the dictation of any group to the others. Engineering education needs to be conceived and organized, not as an isolated form of educational activity, but as an integral part of the professional scheme and the industrial order.

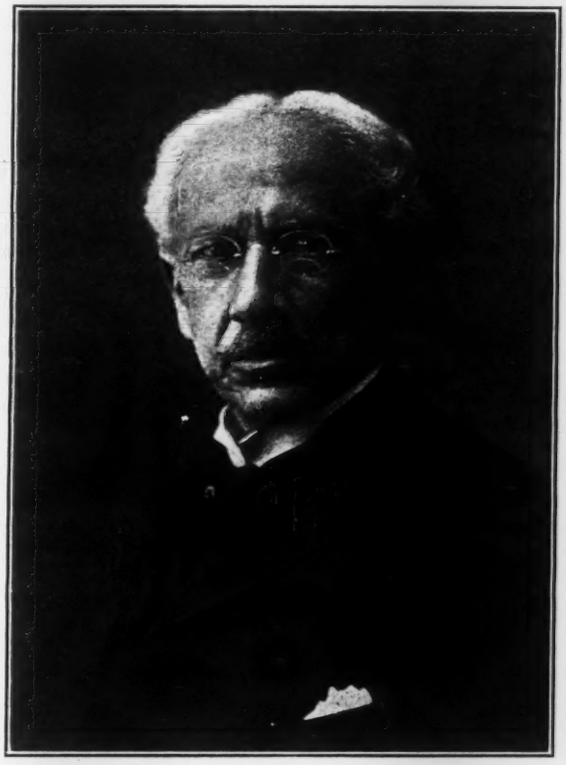
Oberlin Smith Dies

Ninth President of the A.S.M.E. Passes Away at Bridgeton, N. J., at the Age of Eighty-Six

OBERLIN SMITH, President of The American Society of Mechanical Engineers in 1890, passed away early in the morning of Monday, July 19, at his home, Lochwold, Bridgeton, N. J., following an attack of heart failure. He was the last of a notable group of fifteen engineers serving as presidents of the Society between the time of its founding in 1880 and the year 1897. Active up to three days before his death, the eminent inventor and successful manufacturer had passed his eighty-sixth birthday and had rounded out sixty-three years of devotion to a rather unique business career in the little city of Bridgeton.

Born in Cincinnati, Ohio, on March 22, 1840, Oberlin Smith was the son of George R. and Salome (Kemp) Smith, both of whom were of early Colonial English stock. The father was a very active anti-slavery worker and operated a link in that Pre-Civil War institution called the "underground railroad."

When Oberlin Smith was very young the family moved from Cincinnati to Bridgeton, N. J. At Bridgeton the boy began his education in the public schools, and during vacations worked on farms and also learned carpentry. His education was continued at the West Jersey Academy, and his technical training began at the Philadelphia Polytechnic Institute. This latter was of a broad and practical nature. To use his own words in speaking of the period from 1856 to 1861, he ". . . learned drafting, patternmaking, gas and steam fitting, blacksmithing, architectural ironwork, machine work, die making, etc., with schooling mixed in." The "schooling" here referred to was largely home study undertaken on his own initiative.



OBERLIN SMITH

In the early days of the Civil War Mr. Smith became particularly interested in the die working of metals and in the design of dies and presses for this purpose. This interest crystallized in 1863 when he established at Bridgeton a concern for the manufacture of improved dies and presses which he had invented. This business, which in 1877 was incorporated as the Ferracute Machine Company, has been a powerful factor in the commercial development of the die working of metals. Its influence has extended from the minting of Chinese money to the mass production of automotive products.

In the sixty-three years of Oberlin Smith's continuous service as mechanical engineer and president of this concern, he became widely known as an authority on presswork. During this time he designed and built more than five hundred types of presses, and obtained more than fifty patents. For the Atlantic Refining Company and other Standard Oil concerns he worked out in the early eighties can-making devices which were responsible in no small way for the development of the package oil trade with the Far East. In his classic paper Shop Management read before the Society in 1903, Frederick W. Taylor gave Mr. Smith credit for the mnemonic symbol system which is such a characteristic and important feature of Scientific Management.

While most of Mr. Smith's inventions related to presswork, there were several in entirely outside fields. Among them were such widely divergent lines as improved looms, dump carts, keyless locks, and egg boilers. In 1883 he achieved considerable publicity through the invention of a magneto-electric phonograph, and this same publicity prevented him from obtaining a patent upon it. Citizens of Bridgeton bear witness to Mr. Smith's having frequently driven through the streets of that town in a motor-propelled vehicle long before the days of automobiles.

Mr. Smith was a prolific writer and lecturer, his works in this direction covering science, fiction, and even theology. Two of his books on widely different subjects were Press Working of Metals, and Tho Material, Why Not Immortal? His active physical and mental characteristics were reflected in his favorite recreations, which included swimming, rowing, motoring, dancing, and golf, most of which he entered into with enthusiasm to the last of his life.

Some years ago he equipped his phonograph with a mechanism by which records could be changed automatically. In his last years, when his sight was failing, through this means he could sit in his favorite chair and by pressing buttons, with the location of which he had become thoroughly familiar, change the tune being played on an instrument in a nearby room.

His intellectual interests not only covered a very wide range, but he espoused many of them with the utmost vivacity. He was particularly given to astronomical discussions. Mrs. Smith tells of bringing such a debate which Mr. Smith was enjoying with a house guest to an end at one o'clock on a Sunday morning. To her surprise and amusement at six-thirty the next morning she heard the resumption of the heated discussion. Mr. Smith had gotten into bed with his guest and the two were hard at it once more!

It was natural that Oberlin Smith should have been one of the early and active members of The American Society of Mechanical Engineers. He joined the Society in 1881, within a year of its founding. He served as a Manager from 1883 to 1886, and in 1889 succeeded the late Henry R. Towne as the ninth President of the Society. In 1901 his prominence in the engineering field led to his appointment as New Jersey Commissioner to the Pan-American Exposition in Buffalo. He made several European tours for purposes of engineering observation. His acquaintanceship was very wide; among his intimate friends he numbered such well-known national characters as Thomas A. Edison and Henry Ford.

On Christmas Day, 1876, Mr. Smith married Miss Charlotte E. Hill at Bernardston, Mass. He is survived by a daughter, Winifred, Countess Raditchevitch, of Paris, a son, Percival Smith, who is vice-president of the Ferracute Machine Co., and by a brother Fred B. Smith, and a sister, Miss Emily Smith, both of whom live at Lochwold.

Mr. Smith belonged to numerous clubs and societies. In addition to the A.S.M.E., he was a member of the American Institute of Mining and Metallurgical Engineers, the American Institute of Electrical Engineers, the American Society of Civil Engineers, the American Iron and Steel Institute, the Franklin Institute, the American Association for the Advancement of Science, and the Engineers' Clubs of both New York and Philadelphia. He also belonged to the American Automobile Association, the Philadelphia Art Club, The Atlantic Union, the New York Lotus Club, the Luther Burbank Society, the National Academy of Political and Social Science, the National Geographic Society, and the Advisory

Council of the Simplified Spelling Board, and had served as vice-president of the Men's League for Woman Suffrage.

Members of our Society and other friends who had visited Lochwold will probably best remember both Mr. Smith and Mrs. Smith through the quality of their entertaining. This was a home in which gracious hospitality and the very best in American culture were held in high esteem. The Czechs have a saying, "Where there is a guest in the home, there is a God." This might well have been inscribed over the doorway at Lochwold.

The funeral of Oberlin Smith was held at Lochwold at 2:30 on Wednesday afternoon, July 21, the Rev. A. B. Collins of the First Presbyterian Church of Bridgeton, officiating. The American Society of Mechanical Engineers was represented by Fred J. Miller, Past-President, and Morris L. Cooke, Past-Manager, and three Bridgeton members of the Society, Messrs. Henry A. Janvier, Adam C. McCutcheon, and Wilbur R. Smith, who were appointed Honorary Vice-Presidents by President Abbott to represent him.

MORRIS LLEWELLYN COOKE.

Structural-Steel Welding

FOR THE last ten years or more there has been a growing effort to replace riveting by welding in its numerous applications. During the War an attempt was made to produce a rivetless ship, the plates being joined by welding. Of late the effort has been directed in another field, namely building construction. There are obvious advantages of welding as compared with riveting in the erection of steel frames of modern buildings, providing only that at least the same strength can be secured with the same factors of safety and reliability.

From this point of view tests made by A. M. Candy, welding engineer, and G. D. Fish, consulting structural engineer, Westinghouse Electric & Mfg. Co., are of more than usual interest. In this series of tests specimens were all welded with the various members located in the same position and manner as would be required if the various members were actually a part of a building structure. As a result all of the welds were executed from the same direction and were located in the same position as would be found in actual practice. All joints made with the single bead of metal were executed so as to make the bead approximately $^5/_{16}$ in. in thickness at the center and to extend out from the corner of the member so joined a distance of approximately $^5/_{16}$ in.

The amount of weld metal used to make the various joints is unquestionably in excess of that actually required to produce joints of 100 per cent strength, but this was intentionally incorporated in the design of the test specimens so that they could prove beyond the shadow of a doubt that structural members can be joined by arc welding, making joints which will not fail even up to rupturing loads on the members so joined. In carrying out further tests in the future, determinations will be made as to the minimum amount of weld metal which can be used for such joints and still retain the requisite strength at the joints. These tests are claimed to demonstrate, among others, the following facts:

1 The tests demonstrate that welded joints can be constructed in such manner as to develop fully the ultimate strength of the structural members connected.

2 A steel I-beam of given section and length will sustain a far greater load if fixed at its ends by a suitably designed welded joint than if supported by standard riveted connections consisting of top and bottom angles. A 9-in. standard I-beam framed between rigid upright columns 8 ft. apart by means of specially designed welded connections was able to sustain a load 25 per cent greater than a beam of the same size and length framed between columns by means of riveted top and bottom angles of $^1/_2$ in. in thickness.

3 A plate girder assembled by welding and consisting of nothing but sheared plates, has a far greater bending strength than a riveted plate-and-angle girder of the same weight, due to the better distribution of the steel in the cross-section. A 15-in. plate girder which was assembled by welding and simply supported on a 14-ft. span developed more than 50 per cent greater strength than a riveted plate-and-angle girder of the same depth and the same weight.

4 A double-angle tension member such as is used in trusses was connected at the ends by welding, and when tested to tension failure broke through the angles at a load 30 per cent greater than the load at which a hanger consisting of the same size angles with riveted end connections failed.

The joints that were demonstrated in these tests are being used in the five-story steel building 70 by 220 ft., 80 ft. high, now being fabricated and erected by the American Bridge Company for the Sharon Works of the Westinghouse Electric and Manufacturing Company. The typical beam and girder connections are fully continuous connections, thereby permitting large percentage of saving in the weight of the beams and girders to carry a given load. The same type of connection which develops continuity effectively also has the added advantage of stiffening the building against wind loads.

The advantages claimed for welded buildings as compared with the riveted one may be stated as follows:

1 That complete continuity of lines of beams can be obtained in welded construction, whereas, it is well known that this cannot be done in riveted construction.

2 That in a welded building it will be possible to make every joint develop full strength of the main members, whereas, in a riveted building many joints are weaker than the members due to the weakening effects of the rivet holes and the weakness of steel angles which have to be used for transmitting tension between two members at right angles to each other.

At the same time it may be well to recall the objections made with reference to all welded joints by both welding engineers and prospective and actual users of welded structures. Essentially, the main objection is the fact that not only does the personal equation enter with particular importance into all welded work, but thus far no way has been discovered to determine the degree of excellence of a welded joint after it has been made. Because of this there may be quite a margin between what *can* be done by welding and what actually will be done under commercial conditions, with such labor as may be obtainable if welding be applied on that vast scale which would be required in actual building procedure.

Another feature which cannot yet be considered fully established is the degree of reliability of welding made under what might be called adverse conditions; for example, on rusted steel members, in strong wind, and in rainy or sleety weather. However, in all new arts such objections are brought forward and as a rule they have been overcome with further developments. There is no question therefore that such tests as those developed above are of great value in opening new paths of progress, particularly in such a field as building construction which has shown less striking advancement in methods than many other fields of mechanical engineering.

In this connection the following editorial from *Machinery* (London, August 5, 1926) may be of interest as showing the attitude of the British engineering world:

"In spite of the fact that welding is steadily superseding riveting in general engineering work, it should not be thought that riveting is in danger of being entirely supplanted. Anyone who analyzes the subject in an impartial manner will discern that there is wide scope for both methods of jointing.

"Structural engineers, for example, have learned from years of experience the virtues and limitations of rivets and how to use them to make economic structures that will not jeopardize human life. Before they can rely on welding they must adopt standard methods of making welds by different processes. They must determine, and establish values for, the strengths of welded joints of different types. They must determine the grades of structural materials that can be welded satisfactorily and investigate the effects that welding will have on the parent material. Some reliable way should be established of controlling the mechanical and human elements in making a weld and of testing the finished work.

"This is only one example of the urgent need of more detailed and more reliable information on welding technique before the art of welding can be applied to what appears to be an almost unlimited field."

Book Reviews and Library Notes

THE Library is a cooperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

A Handbook of Automotive Standards

S.A.E. HANDBOOK, MARCH, 1926. Published by the Society of Automotive Engineers, New York. Flexible leather, 4½ × 7½ in., 600 pp., diagrams, charts, tables, etc. $5.

MEN give their time for the formulation of standards with the hope, and sometimes with the expectation, that the use of the standards evolved will ultimately become quite universal. The formulation involves a great deal of expense, and this expense is justified only when the resulting standard meets the general approval of the industry and is freely adopted. The use of a standard cannot be forced upon an industry, but if it has merit it will be quickly adopted if the necessary information is put in usable, easily understandable form and, most important of all, made readily available whenever needed.

The S.A.E. Handbook is probably the most successful means used to date to get the standards adopted by an engineering society into the hands of the user, and it is undoubtedly responsible for a large part of the success of the S.A.E. standardization program. The new bound-volume edition should prove even more valuable to the membership, for the user can now be sure that his book is up-to-date and that he is not using a standard that has been changed or has become obsolete. For S.A.E. Standards do change. A too common conception of standardization is the setting of rigid standards and specifications that prohibit all change or progress and limit individuality. Such, however, is not the S.A.E. method. Their standards are continually reviewed in the light of progress, and from time to time such changes are made as become necessary or desirable to keep the standards up to date. Heretofore these changes have been taken care of by issuing new loose-leaf sheets to the membership twice a year, but in the future it is intended to revise and furnish a new bound volume semi-annually.

This book contains the standards and recommended practices of the Society of Automotive Engineers. It also contains some selected general information mostly relating to the standards and recommended practices. While many of the standards and specifications refer to articles of direct interest only to the automotive industry, the majority are being used with equal profit in the machine-tool and many other fields of general machine design. It is inconceivable that a book containing these standards and recommended practices that are reliably estimated to have produced a saving of at least 15 per cent of the total annual retail value of American automotive products, should not be read with interest and used with profit by all mechanical engineers interested in general machine design.

Probably the S.A.E. Standards best known and most used outside of the automotive industry are the specifications and heat treatments for automotive structural steels, now generally known as S.A.E. steels. This list now contains specifications for sixty-three different steels, with recommended heat treatments for most of them, and very conveniently arranged charts giving the physical properties of the more commonly used steels when heat-treated and quenched in either oil or water. The non-ferrous-metal specifications contain recommended mixtures for solders, babbitts, light alloys, and bronzes.

Other standards and recommended practices that can be followed to good advantage in general machine design are the four-, six-, ten-, and sixteen-spline fittings, squared, taper, and serrated shaft fittings, castle nuts, lock washers, flexible steel and non-metallic conduits, storage batteries, insulated cable, cable terminals and clips, ball and roller bearings, tube fittings, brake lining, felt, silent and roller chains and sprockets, flexible-coupling disks, friction-clutch facings, and many articles of minor importance or for occasional use.

The abstract of the report of the Sectional Committee on the Standardization and Unification of Screw Threads giving the necessary information for the use of the new American Standard Screw Threads supplies this information in a very usable form. This standard has been approved as the S.A.E. Standard.

The section on automobile nomenclature is particularly good, and is recommended to other engineering societies as an example of what can be done in a much neglected field.

Some advertising is included at the back of the book, limited, however, to S.A.E. Standard articles and materials. An index of makers of S.A.E. Standard parts and materials is included, but as it is limited to those advertising in the handbook, it is necessarily very meager. It would be very desirable and useful if extended. The chief criticism of the make-up of the book is that the index, being placed in front of the advertising pages, is not readily found.

Standardization of the manner of presentation of standards and of methods of getting the information to the men responsible for putting the standards into use is badly needed, and the S.A.E. Handbook should prove to be a good precedent for other societies to follow.

The automotive industry is to be congratulated on having a society possessed with the foresight to make standardization one of its major activities—with members giving freely of their time for standardization work—and on having the S.A.E. Handbook to get the standards directly into the hands of the users.

J. B. ARMITAGE.[1]

The Investigation of Business Problems

INVESTIGATION OF BUSINESS PROBLEMS: Technique and Procedure. By J. Eigelberner. A. W. Shaw Co., Chicago and New York, 1926. Fabrikoid, 6 × 9 in., 335 pp. $5.

THIS book is a demonstration of the application of the scientific method to the investigation and solution of business problems. The field of investigation is defined and its application to the problems of commerce and industry illustrated. Executive decisions are now based upon special investigations aided by such agencies as commercial research and market surveys, technical and laboratory research, etc. However, good men for economic research are scarce, and there is great need for orderly training of men for this field.

Investigational procedure is divided into five steps: (1) Analyzing the problem; (2) collecting the facts; (3) classifying and tabulating the facts; (4) forming conclusions; (5) establishing the conclusion. The problem must be clearly defined and working hypotheses arrived at so that facts may be collected in an orderly way which

[1] Chief Mechanical Engineer, Kearney & Trecker Corp., West Allis, Wis. Mem. A.SM.E.

will prove the correct hypothesis and cause the others to be discarded, thus forming a logical basis for forming and establishing the correct conclusion.

In addition to professional knowledge and experience, an investigator must have judgment, integrity, poise, vision, perseverance, good physique, and the knack of obtaining the coöperation of those with whom he is associated.

Since "science has derived very little or no benefit from the miscellaneous collecting or grouping of facts without any previous notion of what they are likely to reveal," the author stresses the necessity of a careful preliminary analysis, first getting a broad perspective of the problem and then determining the real objective of the investigation, planning the procedure as regards: (1) The type of investigation; i.e., whether experimental in laboratory or shop, bibliographical, or by various forms of inquiry; (2) the starting point; (3) the subsequent steps; (4) restrictions on the decision; (5) scheduling time and cost, for this is as important in an investigation as in any other type of endeavor; (6) getting the program in writing, which is of great aid to the investigator in keeping him in the proper field of activity; and (7) starting and maintaining a progress record, which provides a permanent written record showing actual accomplishment and which furnishes the basis for measuring accurately the effectiveness of the work.

The investigator must attack his problem with the proper mental attitude, free from prejudice of any sort. He must guard against prejudice occasioned by improper training, custom, or authority. Helpful suggestions for eliminating or controlling the influence of prejudice are given.

Facts may be collected by personal observtion and experiment, through bibliographical research, through interviews, or by questionnaires. In making use of these methods the investigator must develop a knowledge of the mechanics of research work, must have a good memory, be able to take notes effectively, and be able to weigh and test the facts he gathers. In making personal observations and in experimenting it is necessary to cultivate the sensations so that one's impressions are accurate, to learn to concentrate, and to avoid errors of observation.

One chapter discusses the use of bibliographical research, outlining what is available, how it may be obtained, and the best manner of utilizing it. Under the heading of Interviews it is pointed out that the interview is direct and economical; it is the only way certain types of information can be secured, and it permits of a maximum variation in the direction of inquiry. Many valuable instructions are given for the effective use of the interview.

A condensed discussion of the use of questionnaires outlines their advantages and disadvantages, explains how to prepare and secure satisfactory results from them, and gives methods of compiling and utilizing the information thus obtained.

The author states that "jumping at hasty conclusions through failure to suspend judgment until the essential facts have been gathered is one of the principal causes of failure in investigational work." A consideration of the nature of thinking is therefore appropriate. Reflective thinking is the only true type. The close interrelation between concepts, judgments, inference, and reasoning is explained.

Definition of terms used are necessary, and the facts collected must be properly classified. Laws of classification are explained and the use of the recapitulation sheet illustrated. With the facts properly classified it is next necessary to test and properly interpret them. Is there evidence of prejudice or bias? Are the facts pertinent and essential, representative, accurate, and reliable? Is the collection of facts sufficiently comprehensive, and are they comparative?

Under the heading Tabulation and Presentation the purpose, advantages, and technique of tabulation are discussed and the various methods of using averages are explained, attention being called to errors commonly met with in their use.

The systematic and logical manner of forming inferences from the facts, whereby the solution of the problem may be discovered, is explained, and useful rules for developing ideas are given. From these inferences the conclusion is developed. The process of inductive reasoning is carefully considered and its use in finally establishing the conclusion is illustrated.

The final step is the preparation of the report. The elements, functions, and essentials of a good report are clearly described in the closing chapter.

MYRON A. LEE.[1]

Books Received in the Library

BELT CONVEYORS AND BELT ELEVATORS. By Frederic V. Hetzel. Second edition, revised. John Wiley & Sons, New York, 1926. Cloth, 5 × 9 in., 333 pp., tables, diagrams, charts, $5.

The work of an engineer with thirty years of experience in the design and operation of elevators and conveyors, this book is intended as a practical guide to the selection and operation of suitable belt elevating and conveying machinery. The edition has been revised to include the improvements of recent years, particularly in the design of idlers and elevator buckets.

BRASS INDUSTRY IN THE UNITED STATES. By William G. Lathrop. Published by the author, Mt. Carmel, Conn., 1926. Cloth, 5 × 7 in., 174 pp., portraits, 7 × 5 in.

An interesting account of the beginning of this important industry and of its later development. The author traces the early industrial development of Connecticut, shows why brass manufacture became of interest in that locality, and traces the gradual development of the industry and of the important firms engaged in it.

FOUNDRY COST ACCOUNTING. By Robert E. Belt. Second edition. Penton Publishing Co., Cleveland, 1926. Cloth, 6 × 9 in., 267 pp., forms and tables, $6.

The book presents a complete cost-accounting system for each of the principal branches of the foundry industry—the gray iron, the malleable iron, and the steel foundry. The system includes a classification of accounts, methods for distributing overhead expense, and forms showing exactly the sequence of cost entries. It is, the author says, in satisfactory use in successful foundries. The new edition contains new chapters on estimates and quotations, pattern shop costs and estimates, and the distribution of overhead expense during periods of abnormal production.

HYDRAULICS. By Joseph N. Le Conte. McGraw-Hill Book Co., New York, 1926. Cloth, 6 × 9 in., 348 pp., diagrams, tables, $3.

A textbook on the theoretical principles of hydraulics, which directs the attention of the student to first principles for the solution of most problems, instead of to empirical rules or tables. The aim is to teach the student to reason out the basic equations and to master the fundamentals of the subject first by the use of pure mathematics and mechanics.

MATERIALS OF CONSTRUCTION: Their Manufacture and Properties. By Adelbert P. Mills. Third edition, edited by Harrison W. Hayward. John Wiley & Sons, New York, 1926. Cloth, 5 × 9 in., 419 pp., diagrams, charts, $4.

A general textbook, somewhat elementary in character, on the manufacture, properties and uses of the more common materials. The endeavor has been to give a modern treatment in a form concise enough for class use by students of civil engineering. In this edition the text on the constitution of metals, alloy steels and alloys has been expanded, and the chapters on cement, concrete and timber have been revised.

DIE WERKSTOFFE DES MASCHINENBAUES. By A. Thum. Walter de Gruyter & Co., Berlin and Leipzig, 1926. Two vols. Cloth, 4 × 6 in., illus., diagrams, 1.50 r.m. each.

A textbook on the strength and properties of structural materials, written from the viewpoint of their use in mechanical engineering. The first volume discusses the qualities particularly needed in materials for machinery and the extent to which metals exhibit them. It also describes the methods of testing these qualities, and the varieties of iron and steel. In volume 2, the varieties of cast iron, structural steel, cast steel, and non-ferrous alloys are described specifically, and the purposes for which each is suitable are mentioned. A chapter is devoted to such minor machine materials as wood, insulating materials, solders, etc.

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THE ENGINEERING INDEX

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THE ENGINEERING INDEX presents each month, in conveniently classified form, items descriptive of the articles appearing in the current issues of the world's engineering and scientific press of particular interest to mechanical engineers. At the end of the year the monthly installments are combined along with items dealing with civil, electrical, mining and other branches of engineering, and published in book form, this annual volume having regularly appeared since 1906. In the preparation of the Index by the engineering staff of The American Society of Mechanical Engineers some 1200 technical publications received by the Engineering Societies Library (New York) are regularly reviewed, thus bringing the great resources of that library to the entire engineering profession.

Photoprint copies (white printing on a black background) of any of the articles listed in the Index may be obtained at a price of 25 cents a page. When ordering photoprints identify the article by quoting from the Index item: (1) Title of article; (2) Name of periodical in which it appeared; (3) Volume, number, and date of publication of periodical; (4) Page numbers. A remittance of 25 cents a page should accompany the order. Orders should be sent to the Engineering Societies Library, 29 West 39th Street, New York.

ABRASIVE WHEELS

Standardization. Grinding Wheels. U. S. Bur. of Standards—Simplified Practice Recommendation, no. 45, 1926, 23 pp., 33 figs. Study conducted by Bur. of Standards resulted in reduction from 715,200 to 255,800 varieties of grinding wheels.

ACCIDENT PREVENTION

Machine Shops. Reducing Lost Time Accidents, S. DeHart. Am. Mach., vol. 64, no. 25, June 24, 1926, pp. 967–969, 6 figs. Studying shop accidents in detail; slight effort reduces number and severity of accidents; best methods of prevention; practical results in time and monetary savings.

ACCIDENTS

Industrial. Accidents Common to Industry, W. F. Dittmer. Safety Eng., vol. 51, no. 6, June 1926, pp. 328–332, 3 figs. Discusses accident problem at East Pittsburgh Works of Westinghouse Elec. & Mfg. Co.; classification of accidents for 1924 and 1925; reduction of eye accidents.

The Human Factor and Industrial Accidents, H. M. Vernon. Int. Labour Rev., vol. 13, no. 5, May 1926, pp. 673–683. Examination of factors concerned in accident causation with view to improving preventive methods, shows that very much larger proportions of industrial accidents are due to carelessness or inattention, lack of knowledge or lack of skill on part of human subject involved than to dangerous conditions under which work is done; discussion of main causes of carelessness and inattention, atmospheric and other conditions which render worker more liable to accident, and effects of experience and age.

AERODYNAMICS

Wind-Tunnel Research. The Resistance of the Airship Models Measured in the Wind Tunnels of Japan. Tôkyô Imperial Univ.—Report of Aeronautical Research Inst., vol. 2, no. 15, Mar. 1926, 84 pp., 28 figs. Report of investigation to compare resistance or resistance coefficient of airship models measured at different wind tunnels and by different methods to deduce some properties, if any, peculiar to wind tunnels and aerodynamical balances employed.

AIR COMPRESSORS

Care and Operation. Routine Care and Operation of Air Compressors. Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 842–845, 3 figs. Air compressors need clean air, proper cooling, and high-grade mineral oil in cylinders; proper lubrication of all bearings also important.

AIRCRAFT

Defense Against. The Past and Future of Defense Against Aircraft, B. F. Harmon. Coast Artillery Jl., vol. 63, nos. 5 and 6, Nov. and Dec. 1925, pp. 449–460 and 555–566. Shows great importance of roles played by Air Service; illustrates importance of Air Service roles, particularly reconnaissance, by reference to opening phases of World War; anti-aircraft means of defence; etc.

Steam Power Plants for. Steam Power Plants in Aircraft, E. E. Wilson. Nat. Advisory Committee for Aeronautics—Tech. Notes, no. 239, June 1926, 30 pp., 2 figs. Analysis of proposal to use steam power plants in aircraft.

AIRCRAFT CONSTRUCTION MATERIALS

Compression Rubbers. Compression Rubbers, G. H. Dowty. Flight (Aircraft Engr.), vol. 18, no. 25, June 24, 1926, pp. 60–62, 7 figs. Properties of compression rubbers and results of tests of various types of compression rubbers.

AIRPLANE ENGINES

Air-Cooled. Adaptation of the Radial Air-Cooled Engine, R. W. A. Brewer. Aviation, vol. 20, no. 25, June 21, 1926, pp. 942–944, 12 figs. Special problems of installation; cowling and cooling; materials for manifolds.

Navy Air-Cooled Engines Development, E. E. Wilson. Aviation, vol. 21, no. 2, July 12, 1926, pp. 59–61, 3 figs. Weight per horsepower of air-cooled engines; fuel consumption favors water-cooled engines; Navy's program for development of air-cooled engines.

Curtiss. The Curtiss D-12 Engine in Great Britain. Aviation, vol. 20, no. 25, June 21, 1926, p. 945, 1 fig. Engine passes Air Ministry 100-hr. test and is accepted and approved for use in British military aircraft.

Fuel-Injection Valves. An Investigation of the Characteristics of Steel Diaphragms for Automatic Fuel-Injection Valves, W. F. Joachim. Nat. Advisory Committee for Aeronautics—Tech. Notes, no. 234, Apr. 1926, 22 pp., 15 figs. Research on steel diaphragms undertaken at Langley Memorial Aeronautical Laboratory as part of general investigation of fuel-injection engines for aircraft; work determined load-deflection, load-deformation and hysteresis characteristics for single diaphragms having thickness from 0.002 to 0.012 in. and for similar diaphragms tested in multiple having total thicknesses from 0.012 to 0.18 in.

Radiators. Aircraft Radiators. Automobile Engr., vol. 16, no. 216, June 1926, pp. 220–222, 10 figs. Recent developments with long-tube honeycomb type.

Valve Seats, Machining. Machining Valve Seats in an Aeronautical Engine, G. A. Tilsher. Am. Mach., vol. 65, no. 1, July 1, 1926, pp. 21–23, 8 figs. Features involved in design of tools for each operation; single fixture and one machine unit suffices.

AIRPLANE PROPELLERS

Design. Propeller Design—A Simple System Based on Model Propeller Test Data, F. E. Weick. Nat. Advisory Committee for Aeronautics—Tech. Notes, no. 237, May 1926, 19 pp., 12 figs. Simple system for designing propellers of standard form; based on tests of 13 Navy models; method has given satisfactory results in Bur. of Aeronautics, U. S. Navy Department.

Propeller Design—Extension of Test Data on a Family of Model Propellers by Means of the Modified Blade Element Theory, F. E. Weick. Nat. Advisory Committee for Aeronautics—Tech. Notes, no. 236, May 1926, 8 pp., 6 figs. Describes method to extend data obtained from tests on 13 model propellers to include all propellers of same form likely to be met in practice.

Theory. Theory of Airscrews, S. Kawada. Tôkyô Imperial Univ. of Aeronautical Research Inst.—Report, vol. 1, no. 14, Mar. 1926, pp. 361–404, 8 figs. Attempts to elucidate mechanism of action of airplane propeller in air, based on Prandtl's theory of airfoil.

AIRPLANES

Arresters and Catapults. Airplane Arresters and Catapults, H. C. Pratt. Aviation, vol. 2[illegible] no. 4, July 26, 1926, pp. 122–124. Possibility of f[illegible]e use in commercial aviation.

Commercial, 3-Engined. The Armstrong-Whitworth "Argosy." Flight, vol. 18, no. 914, July 1, 1926, pp. 368–369, 3 figs. Latest 3-engined commercial airplane is fitted with three Siddeley Jaguar engines developing total of nearly 1200 hp.; accommodates 20 passengers and carries sufficient gas for flight of 400 miles; normal cruising speed 90 to 95 miles per hour.

Control at Low Speeds. On the Control of Airplanes at Low Speeds, B. V. Korvin-Kroukovsky. Aviation, vol. 20, no. 25, June 21, 1926, pp. 946–947, 3 figs. Supporting existing control methods as adequate for control under all conditions.

Curtiss. The Curtiss Pursuit Training Plane. Aviation, vol. 21, no. 1, July 5, 1926, pp. 24–25, 2 figs. New advanced plane with Wright E-2 engine; preliminary flight tests show high speed of 132.8 miles per hour with landing speed of 54.6 miles per hour; construction economical since all parts except power plant are identical with regular Hawk plane which has been in production for over a year.

De Havilland. A New De Havilland Commercial Aeroplane. Flight, vol. 18, no. 23, June 10, 1926, pp. 329–333, 9 figs. Particulars of D. H. 66 with three Bristol Jupiter engines.

Flying Boats. See FLYING BOATS.

Laird. The Laird Commercial. Aviation, vol. 21, no. 4, July 26, 1926, pp. 125–126, 2 figs. Three-seater commercial and touring type fitted with either OX-5, C-6, or Whirlwind engine.

Line Assembly. Line Assembly of Airplanes at the Breguet Works. (La fabrication des avions "à la chaine" aux usines Breguet), Aéronautique, vol. 8, no. 85, June 1926, pp. 193–196, 6 figs. Description of methods employed in vertical erection of wings and assembly of fuselages and frames of Breguet 19 metal biplane, with routing diagrams.

Metal. Metal Aeroplane Construction, A. B. Miller, Junior Instn. Engrs., vol. 36, May 1926, pp. 325–343, 17 figs. Review of construction at present stage of selective process which continues its development due to search for light and permanent structure, capable of production and maintenance, in same way as any other manufactured article, by engineering industry; factors leading to choice of metals as most suitable construction materials are: weight economy, lack of deterioration of spares in storage, good supply of raw material, and facilities for mass production and standardization.

R.A.F. Display. On the Royal Air Force Display. Aeroplane, vol. 31, no. 1, July 17, 1926, pp. 1–3, 8 figs. Description of R.A.F. display, at Hendon; record of events and comments of airplanes seen.

Ryan. New Ryan Monoplane Permits Change of Engines in 20 Minutes, A. F. Denham. Automotive Industries, vol. 55, no. 5, July 29, 1926, pp. 174–175, 3 figs. Entire nose of ship, including motor, demountable; wing is attached directly to structural part of fuselage; weight of plane fully loaded is 1750 lb.

Seaplanes. See SEAPLANES.

Spar and Wing-Rib Tests. Comparison of Tests on Experimental 15-Inch Metal Spars and 11-Foot Chord Metal Wing Ribs, J. S. Newell. Air Service Information Circular, vol. 6, no. 556, Mar. 1, 1926, 43 pp., 54 figs. Results and conclusions of tests made at McCook Field.

Static Test of Kerber-Boulton Experimental Spar, E. R. Weaver. Air Service Information Circular, vol. 6, no. 557, Mar. 1, 1926, 7 pp., 12 figs. Results of test of experimental Warren truss-type spar built up of duralumin and steel tubes.

Velocity. The Effect of Flight Path Inclination on Airplane Velocity, W. S. Diehl. Nat. Advisory Com-

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NOTE.—The abbreviations used in indexing are as follows:

Academy (Acad.)
American (Am.)
Associated (Assoc.)
Association (Assn.)
Bulletin (Bul.)
Bureau (Bur.)
Canadian (Can.)
Chemical or Chemistry (Chem.)
Electrical or Electric (Elec.)
Electrician (Elecn.)
Engineer (Engr.[s])
Engineering (Eng.)
Gazette (Gaz.)
General (Gen.)
Geological (Geol.)
Heating (Heat.)
Industrial (Indus.)
Institute (Inst.)
Institution (Instn.)
International (Int.)
Journal (Jl.)
London (Lond.)
Machinery (Machy.)
Machinist (Mach.)
Magazine (Mag.)
Marine (Mar.)
Materials (Matls.)
Mechanical (Mech.)
Metallurgical (Met.)
Mining (Min.)
Municipal (Mun.)
National (Nat.)
New England (N. E.)
Proceedings (Proc.)
Record (Rec.)
Refrigerating (Refrig.)
Review (Rev.)
Railway (Ry.)
Scientific or Science (Sci.)
Society (Soc.)
State names (Ill., Minn., etc.)
Supplement (Supp.)
Transactions (Trans.)
United States (U. S.)
Ventilating (Vent.)
Western (West.)

mittee for Aeronautics—Report, no. 238, 1926, 11 pp., 6 figs. Systematic study of relations between flight velocity and its horizontal component in power glides; effect of inclination of flight path on horizontal velocity; effect of slight deviation from horizontal flight path.

Wing Flutter. Wing Flutter of Monoplanes (Flügelschwingungen an freitragenden Eindeckern), A. Raab. Zeit. für Flugtechnik u. Motorluftschiffahrt, vol. 17, no. 7, Apr. 14, 1926, pp. 146–147. Wings of monoplanes with little or no external bracing have been found to flutter especially when subjected to high speed or to gusts; this condition becomes critical if flier does not know its cause but tries to land rapidly; complete collapse from this cause has occurred; wings of internally trussed monoplanes should be made of light and strong materials without too heavy connecting pieces.

AIRSHIPS

Electric Hazards. Electric Hazards of Airship Traffic (Die elektrischen Gefahren des Luftschiffverkehrs), Wigand. Zeit. für Technische Physik, vol. 7, no. 5, 1926, pp. 238–239. Discusses formation of strong electric fields and electric sparks, electrostatic field in atmosphere, electric charge of airships, etc., and shows how dangers may be eliminated.

AIRWAYS

Meteorological Conditions. Meteorological Conditions along Airways, W. R. Gregg. U. S. Natl. Advisory Comm. for Aeronautics, Report No. 245, 1926, 16 pp., 8 figs. Chicago-Fort Worth airway studied to show kind of meteorological information needed to determine operating conditions along airways.

ALLOYS

Aluminum. See ALUMINUM ALLOYS.

Aluminum Bronze. See ALUMINUM BRONZE.

Gun Metal. See GUN METAL.

Iron. See IRON ALLOYS.

Nickel. See NICKEL ALLOYS.

ALUMINUM ALLOYS

Duralumin. See DURALUMIN.

Refining Processes. Processes of Refinement in Aluminum Alloys (Die Veredelungsvorgänge in vergütbaren Aluminiumlegierungen), K. L. Meissner. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 12, Mar. 20, 1926, pp. 391–401, 13 figs. Nature of automatic refinement of duralumin; explanation of refining phenomena according to Merica; tests by Fraenkel on electric conductivity and chemical resistance; influence of age hardening at room and at higher temperatures; effect of compound Mg_2Si; refining process according to Hanson and Gayler, Hond and Konno; refining of magnesium-free aluminum containing copper by artificial age hardening; peculiarities of magnesium and magnesium-free alloys when used in light-metal construction; tests with lautal; critical dispersion.

ALUMINUM BRONZES

Heusler Ferromagnetic. Crystal Structure of Heusler Alloys (Ueber die Kristallstruktur der Heuslerschen Legierungen), L. Harang. Physokalische Zeit., vol. 27, no. 7, Apr. 1, 1926, pp. 204–205, 1 fig. Determination of structure of Heusler ferromagnetic aluminum-manganese bronzes; there is no simple relationship between magnetic properties and appearances of different lattices.

AMMONIA COMPRESSORS

Design. Some Notes on Ammonia Compressor Design, K. Lightfoot. Cold Storage, vol. 29, no. 339, June 17, 1926, pp. 252–254, 3 figs. Double-acting horizontal vs. single-acting vertical.

Troubles, Prevention of. Preventing Ammonia Compressor Troubles, H. J. Macintire. Power Plant Eng., vol. 30, no. 12, June 15, 1926, pp. 708–710, 1 fig. Discusses factors that may reduce efficiency of compressor, such as incorrect pressures, non-condensable gases and poor heat transfer in system.

AMMONIA CONDENSERS

Operation. Operating Characteristics of Ammonia Condensers, W. H. Motz. Power, vol. 63, no. 26, June 29, 1926, pp. 1012–1014, 7 figs. Influence upon condenser pressure of initial temperature and quantity of cooling water, amount of surface, and disposition of surface; condenser pressures for various conditions and effect of reduction of condenser pressure on power consumption.

APPRENTICES, TRAINING OF

Tractor Manufacturing Plant. Apprentice Training System of the Caterpillar Tractor Co., San Leandro, California. West. Machy. World, vol. 17, no. 6, June 1926, pp. 237–239, 5 figs. System employed by Caterpillar Tractor Co., Cal., giving shop training and classroom instructions in 4-year course with payment during training.

AUTOMOBILE ENGINES

Crankcase-Oil Dilution. Oil Rectification. Automobile Engr., vol. 16, no. 216, June 1926, pp. 226–228, 5 figs. Details of device for preventing crankcase dilution.

Fiat. The Fiat Two-Stroke Racing Engine. Automotive Industries, vol. 54, no. 24, June 17, 1926, p. 1046, 1 fig. Engine comprises six vertical cylinders arranged in line, each containing two pistons; racing engine is provided with centrifugal supercharger and exhaust through ports in cylinder wall at end of stroke.

Fuels. See AUTOMOTIVE FUELS.

Maedler. Six-Cylinder, Two-Stroke Oil Engine Built for Bus Service, P. M. Heldt. Automotive Industries, vol. 54, no. 24, June 17, 1926, pp. 1042–1044, 3 figs. Double-piston type designed to develop 125 hp. at 1200 r.p.m.; has $3^1/_2$-in. bore and $7^1/_4$-in. stroke; intake and exhaust ports unusually large; developed by Maedler Engine Corp., Cleveland, O.

Manufacture. The Works of W. H. Dorman & Co., Ltd. Automobile Engr., vol. 16, no. 216, June 1926, pp. 209–213, 12 figs. Methods and equipment employed in production of engines.

Oil Filtration. Improved Engine Oil Filtration. Motor Transport, vol. 42, no. 1109, June 14, 1926, pp. 670–671, 4 figs. Describes three types of special oil filters which, it is claimed, yield more mileage for less consumption and wear: absorbent-pad type, Skinner combined filter and recovery unit, and Renault oil purifier.

Supercharging. Supercharging Increases Power Output, Lowers Fuel Efficiency, C. B. Dicksee. Automotive Industries, vol. 54, no. 24, June 17, 1926, pp. 1048–1053, 8 figs. When used to compensate for loss of volumetric efficiency due to high speed of engine or high altitude, substantial increase in output is gained for slight power absorption.

AUTOMOBILES

Bentley. The Three-Litre (15.9 H.P.) Bentley. Auto-Motor Jl., vol. 31, no. 25, July 8, 1926, pp. 513–515, 7 figs. Description of 15.9-hp. car and its mechanical details.

Buick. Buick 1927 Line Has Counter-Balanced Crankshaft and New Transmission, L. S. Gillette. Automotive Industries, vol. 55, no. 5, July 29, 1926, pp. 163–165, 3 figs. Other mechanical additions include crankcase-ventilating system, rubber-cushioned power plant and AC oil filter.

Chassis Lubricator. Chassis Lubricator Designed to Feed Oil Automatically as Needed. Automotive Industries, vol. 54, no. 25, June 24, 1926, p. 1103, 2 figs. Connected with springs so that major deflections, as occur on rough roads, cause oil to be forced through system.

Donnet-Zedel. The 10 HP. Donnet-Zedel. Auto-Motor Jl., vol. 31, no. 23, June 24, 1926, pp. 477–479, 10 figs. Two new models of 10-hp. Donnet Zedel automobile which combine fine design and engineering with light weight and high efficiency with reasonable first cost and running costs.

Gwynne. The New Gwynne. Auto-Motor Jl., vol. 31, no. 24, July 1, 1926, pp. 503–504, 7 figs. Model has monobloc engine with overhead valves; 57 mm. bore and 100 mm. stroke; valve springs have been increased in strength, cam contours and engine crankshaft redesigned, and other changes made increasing efficiency and giving greater comfort.

Headlights. New Headlight Has Long Range Without Glare, M. A. Hall. Automotive Mfr., vol. 68, no. 3, June 1926, pp. 5–7, 3 figs. Headlight demonstrated by Gen. Elec. Co. at Schenectady which has long range without glare; wide side beam with backward rays which illuminates ditches and road signs and requires no focusing.

Photometric Measurements and Glare Tests of Automobile Headlights (Misure forometriche e di abbagliamento in fari per autoveicoli). E. Perucca. Elettrotecnica, vol. 13, no. 3, Jan. 25, 1926, pp. 45–56, 11 figs. Suggested tests for automobile headlamps; convenient method for halving distance required for photometric measurements by means of reflection by plane mirror; describes several headlights and give results obtained by testing them in manner suggested.

Itala. Front Drive Itala Racers. Autocar, vol. 56, no. 1599, June 25, 1926, pp. 1023–1024, 3 figs. New 12-cylinder 1100 and 1500-cc. productions; engine speeds run up to 8000 r.p.m.

Itala Builds 67 Cu. In. Racer with 12 Cylinders and Front Drive, W. F. Bradley. Automotive Industries, vol. 55, no. 3, July 15, 1926, pp. 96–97, 1 fig. Engine has speed of 8000 r.p.m. and 100 m.p.h. is claimed at 6000 r.p.m.: Roots blower used for supercharger.

Lanchester. The 21 H.P. Lanchester Chassis. Automobile Engr., vol. 16, no. 216, June 1926, pp. 202–207, 12 figs. Designed to accommodate large enclosed bodies, but at same time to be sufficiently lively to make good open car, fitted with ordinary 4-speed sliding gear box instead of epicyclic gear used on 40-hp. model; has 6-cylinder engine in unit with clutch and gear box and enclosed propeller-shaft and worm drive.

Mudguard Construction. Making Mudguards with Hand Tools. Machy. (Lond.), vol. 28, no. 717, July 8, 1926, pp. 389–391, 9 figs. Layouts for new guards; making wood forming blocks; preparing blank; hammering work to shape; trimming, smoothing, and polishing, fitting and assembling.

Oldsmobile. Oldsmobile Engine is Made Larger; New Body Model Added, L. S. Gillette. Automotive Industries, vol. 55, no. 3, July 15, 1926, pp. 86–88, 5 figs. Cylinder bore increased, adding 16 cu. in. to piston displacement; crankcase ventilating system; oil and air cleaners; harmonic balancer

Overland. New Overland "Whippet" America's Smallest Car, L. S. Gillette. Automotive Industries, vol. 54, no. 25, June 24, 1926, pp. 1084–1088, 10 figs. Has 134-cu. in. engine with rated horsepower of 15.6 but developing maximum of 30 hp. at 2800 r.p.m.; four-wheel brakes, 27-in. tires and semi-elliptic springs.

Steering Gear. Dual Steering Gear Designed to Cure Wheel "Shimmy," W. F. Bradley. Automotive Industries, vol. 55, no. 5, July 29, 1926, pp. 170–171, 5 figs. Each wheel is controlled separately in system developed by Waseige of Farman Automobile Co., France; claimed to eliminate transverse oscillations of radiator and hood.

Swift. The New 12-35 Swift. Auto-Motor Jl., vol. 31, no. 21, June 10, 1926, pp. 435–437, 10 figs. Equipped with engine which will develop over 35 hp.; special 4-speed silent gear; 4-wheel brakes.

Talbot. New $91^1/_2$ In. Talbot Racer Designed for European Tracks, W. F. Bradley. Automotive Industries, vol. 54, no. 24, June 17, 1926, pp. 1054–1055, 4 figs. French firm develops new models for coming Grand Prix races; girder frame used; front axle made in halves and bolted together at middle; equipped with Roots blower.

AUTOMOTIVE FUELS

Anti-Knock. Motyl and Motalin (Motyl und Motalin), Ostwald. Petroleum, vol. 22, no. 14, May 10, 1926, pp. 525–530, 8 figs. Discusses low volatility, impurities and knocking of gasoline; production of motalin, dapolin colored yellow by addition 4 per cent of motyl, which prevents knocking, iron carbonyl replacing dangerous tetraethyl lead.

France. Better Use of Liquid Fuels (Sur la meilleure utilisation des combustibles liquides), P. Dumanois. Annales des Combustibles Liquides, vol. 1, no. 1, 1926, pp. 49–78. Increasing efficiency of explosion engines by increasing compression, and consequent appearance of spontaneous ignition and detonation; anti-knock compounds and possibility of using heavier oil.

AVIATION

Air-Mail Service. Elko, Nev.–Pasco, Wash., Air-Mail, C. T. Wrightson. Aviation, vol. 20, no. 26, June 28, 1926, pp. 978–979, 2 figs. Varney air-mail route, C.A.M. 5, now operating regularly with Whirlwind-engined Swallow air-mail planes.

Civil, France. French Civil Aviation. Aviation, vol. 21, no. 4, July 26, 1926, pp. 127–129, 4 figs. Review of French air transportation with statistics for past three years of operation.

B

BEARINGS

Babbitted. Babbitting Bearings. Mech. World, vol. 79, no. 2060, June 25, 1926, pp. 489–490, 8 figs. Discusses method for babbitting bearings; removing old babbitt; housing; tools for cutting oil grooves, etc.

Motor, Lubrication of. Lubricating Motor Bearings, F. E. Gooding. Indus. Engr., vol. 84, no. 7, July 1926, pp. 311–315, 10 figs. Problems in lubricating motor bearings and methods used by manufactures of motors and bearings to reduce operating troubles.

BEARINGS, BALL

Calculation. Construction of Ball Bearings (Ueber die Konstruktion von Rillenkugellagern), A. Palmgren. Maschinenbau, vol. 5, no. 10, May 20, 1926, pp. 452–455, 6 figs. Discusses bearings derived from Stribeck type; effect of contact between balls and race on load capacity; shows by calculations that bearings without filling groove and smaller number of balls are preferable.

Journal-Thrust. Journal-Thrust Ball Bearings. Automobile Engr., vol. 16, no. 216, June 1926, p. 208, 8 figs. Combination journal and thrust bearing somewhat similar to French duplex bearing; another type is known as U or Universal bearing, in which advantage is taken of thrust-resisting feature to build up effective double-journal and double-thrust bearing.

BELT DRIVE

Individual vs. Group. Group Drives More Economical for Most Conditions, L. H. Hopkins. Belting, vol. 28, no. 6, June 1926, pp. 15–17, 1 fig. Group drives show greater efficiency from standpoint of operation, maintenance and initial cost and are more adaptable to slight speed changes.

Small. Small Belt Drives, J. M. Combs. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 934–937, 8 figs. Their general application, types of belts used and design of efficient pulleys.

BLAST FURNACES

Drying of Blast. The Drying of Blast by Means of Silica-Gel (Das Trocknen des Geblasewindes durch Silica-Gel), F. Krull. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 27, July 3, 1926, pp. 907–910, 2 figs. Points out advantages accruing from use of dry blast and describes silica-gel process, and its practical application, with cost data.

BOILER FEEDWATER

Deaeration. Importance of Care of Feedwater in Economic Boiler Operation (Die Bedeutung der Speisewasserpflege für den wirtschaftliches Dampfkesselbetrieb), H. Riemer. Sparwirtschaft, nos. 4 and 5, Apr. and May 1926, pp. W55–W60 and W76–W81, 18 figs. Formation of boiler scale, solubility of gases in water, deleterious effect of organic compounds. Physical and chemical processes of feedwater purification.

Softening. The Why and How of Water Softening, S. O. Andros. Steam Power, vol. 5, no. 5, June 1926, pp. 3, 5 and 12, 4 figs. Gives case representing typical condition arising from use of raw water and water-softening method employed.

Treatment. The Treatment of Boiler Feed Water, D. Brownlie. Indus. Chemist, vol. 2, no. 17, June 1926, pp. 274–275. Methods of treatment without use of softening plant.

BOILER FIRING

Schuekert-Petri System. Schuekert-Petri Method of Powdered Fuel and Grate Firing. Nat. Engr., vol. 30, no. 7, July 1926, pp. 291–295, 4 figs. Operating methods and results obtained by firing boilers with combination of powdered fuel and traveling grate, in which powdered fuel is burned as auxiliary process in ordinary traveling-grate furnace without ignition arch

BOILER FURNACES

Air Preheating. A New Air Preheater. Eng. & Boiler House Rev., vol. 39, no. 11, May 1926, pp.

520–524, 2 figs. General principle of new Perry design which, it is claimed, is considerably less in weight and bulk than existing types of preheaters or economizers, while for high duties, advantages, together with low cost, are increased.

Preheating Combustion Air in Boiler Firing (Ueber die Vorwärmung der Verbrennungsluft bei Dampfkesselfeuerungen), E. Mikocki. Sparwirtschaft, vol. 3, no. 5, May 1926, pp. W73–W76. Discusses recent tests made and advantages of air preheating; types of preheaters, including that of Schwabach and Ljungstrom.

Utilization of the Waste Heat of Boiler Installations to Preheat the Combustion Air, E. Blau. Combustion, vol. 14, no. 6, June 1926, pp. 374–377, 2 figs. By preheating combustion air, 70 per cent of heat of flue gases can be recovered; temperature in combustion chamber is increased appreciably, and consequently combustion on grate proceeds more vigorously; system of air heaters in common use. Translated from German.

Automatic Regulation. Automatic Furnace Regulation in Boiler Operation (Selbsstätige Feuerungsregelung im Kesselbetrieb), E. Schulz. Zeit. des Vereines deutscher Ingenieure, vol. 70, nos. 21 and 25, May 22 and June 19, 1926, pp. 718–724 and 855–859, 32 figs. Describes arrangements and equipment of various American and European firms, general requirements for separate boilers and boiler sets; governors for control of air and for whole boiler plant; electric and mechanical action.

Calculation. Furnaces for Boiler Plants. (Feuerungen für Kesselanlagen), H. E. Witz. Archiv für Warmewirtschaft, vol. 7, no. 6, June 1926, pp. 169–172, 10 figs. Calculation of heat radiated from grate to boiler; comparison of radiation to water-tube boiler from grate and from pulverized-coal firing; new design of pulverized-coal furnace.

Design. Progress in Furnace Construction (Fortschritte im Feuerungsbau), H. Wellmann. Wärme- u. Kälte-Technik, vol. 28, no. 8, Apr. 20, 1926, pp. 79–82, 7 figs. Describes Vervoort type of traveling grate 4000 mm. in width, with 22-sq. m. grate surface, built by Vervoort Wanderrost A. G., Düsseldorf; advantages of wide grates over double grates.

Excess-Air Reduction. Limiting Factors in Reducing Excess Air in Boiler Furnaces, E. G. Bailey. Mech. Eng., vol. 48, no. 7, July 1926, pp. 703–709, 16 figs. Study of interrelationship and relative importance of three factors most closely related to excess air; namely, furnace temperature, unburned fuel, and heat loss in chimney gases.

Oil-Fired. Good Design Increases Life of Furnace, F. A. Rothwell. Power Plant Eng., vol. 30, no. 13, July 1, 1926, pp. 740–743, 6 figs. Changes which were made in setting of oil-fired boiler and benefits resulting; air cooling of walls adds to life and efficiency.

BOILER OPERATION

Control. Regulating Devices and Measuring Instruments in Boiler Plants (Neue Regeleinrichtungen und Messgeräte für Dampfkesselanlagen), Elektrotechnik u. Maschinenbau, vol. 44, no. 12, Mar. 21, 1926, pp. 60–63, 5 figs. Roncka automatic regulator known as multimeter, which controls velocity of combustion, air supply, feedwater supply, fuel supply, etc.; examples of applications.

Temperature Measurement. Temperature Measurement in the Boiler House, D. Brownlie. Eng. & Boiler House Rev., vol. 39, no. 12, June 1926, pp. 568–572, 4 figs. Discussion of instruments employed.

BOILER PLATE

Caustic Embrittlement of. Caustic Embrittlement, S. W. Parr and F. G. Straub. Power, vol. 63, no. 26, June 29, 1926, pp. 994–998, 10 figs. Summary of results of studies carried on at engineering experiment stations of Univ. of Illinois to discover process whereby seemingly good boiler plate becomes worthless after few years of service.

BOILERS

American, Superiority of. Superiority of the American Boiler. Iron Age, vol. 118, no. 6, Aug. 5, 1926, pp. 346–347. Review of address to Society of German Engineers by F. Von Muenzinger, analyzing underlying differences in Germany and United States; superiority is said to be specially evident in construction of boiler accessories; execution of large forced-draft furnaces, chain grates, underfeed stokers, devices for removal of ashes, automatic feeding and construction of brickwork for large boilers.

Coke-Fired. Coke Firing for Steam Boilers, E. W. L. Nicol. Eng. & Boiler House Rev., vol. 39, no. 12, June 1926, pp. 563–565, 2 figs. Advantages of blending; analysis of fuels used by means of sandwich system.

Economy, Determination of. Determining Boiler Economy, D. Cochrane. Power House, vol. 19, no. 12, June 20, 1926, pp. 23–24. Determination of economy in coal consumption, in weight of boiler, in first cost, and in maintenance of total life.

Heat Transfer. Heat Transfer in Boilers, C. F. Wade. Elec. Times, vol. 69, no. 1808, June 17, 1926, pp. 705–707, 4 figs. Trend of design of steam boilers; transfer of heat by radiation and by conduction aided by convection.

Locomotive. See LOCOMOTIVE BOILERS.

Marine. See MARINE BOILERS.

Therm as Unit of Capacity. The "Therm" as a Unit of Boiler Capacity, H. W. Brooks. Combustion, vol. 14, no. 6, June 1926, p. 377. It is believed that therm would lend itself ideally to solution of problem.

Water-Level Gages. The "Simplex" Distant Water Level Indicator. Eng. & Boiler House Rev., vol. 39, no. 12, June 1926, pp. 581–582, 2 figs. Simplex distant water-level indicator consists of float chamber, which is connected to steam and water space of boiler; indicating rod is suspended from float by means of chain and reciprocates in gage proper.

Water-Tube. See BOILERS, WATER-TUBE.

BOILERS, WATER-TUBE

Sulzer. The Sulzer Water-Tube Boiler. Eng. & Boiler House Rev., vol. 39, no. 12, June 1926, p. 578, 1 fig. Principal claims are those of simplicity in construction and design which lends itself to requirements of modern conditions for high pressures and high steaming capacity; consists of large steam and water drums connected by vertical or nearly vertical water tubes thus providing for rapid circulation and large combustion chamber of volume.

BORING MACHINES

Electric Control. Electric Control Equipment for Large Boring Mill, W. E. Ross. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 965–966, 2 figs. Equipment installed at Peterboro plant of Canadian General Electric Co. said to be one of the largest of its kind in world.

Stern-Bush. Stern-Bush Boring Machine. Engineering, vol. 121, no. 3152, June 11, 1926, pp. 692, 8 figs. Details of machine for rebushing stern-tube liner of vessel in position.

BRAKES

Specifications. Brakes and Brake Equipment. Ry. Age, vol. 80, no. 32, June 12, 1926, pp. 1732–1735, 5 figs. Report of committee before Am. Ry. Assn. See also Ry. Rev., 78, no. 25, June 19, 1926, pp. 1152–1154.

Temperature Stresses in Wheels. Temperature Stresses in Brake Wheels, A. J. S. Pippard. Engineering, vol. 122, no. 3158, July 23, 1926, pp. 93–94. Discusses temperature-stress effect in case of brake applied direct to rim of flywheel having straight spokes.

BUSES

Trolley. Departmental Trolley Buses of Gard (Les omnibus à trolley départmentaux du Gard), Génie Civil, vol. 88, no. 14, Apr. 3, 1926, pp. 309–315, 22 figs. Describes two lines between Nimes and Gard (France) of electric buses operated with direct current of 550 to 600 volt from 4 substations; design and electric equipment of buses; also used with trailers and for freight service.

C

CABLEWAYS

Funicular Lines. Safety Devices in Cableways for Passenger Traffic (Dispositif de sécurité dans les funiculaires aériens pour voyageurs), F. Crestin. Génie Civil, vol. 88, no. 22, May 29, 1926, pp. 482–485, 14 figs. Discusses principal causes of cable deterioration; safety devices, brakes, etc., and describes Mont Blanc line and its equipment.

CAMS

Grinding. Cam Grinding, H. Darbyshire. Automobile Engr., vol. 16, no. 216, June 1926, p. 219, 1 fig. Considerations regarding economical production of cams.

CAR COUPLERS

Specifications. Couplers and Draft Gears. Ry. Rev., vol. 78, no. 25, June 19, 1926, pp. 1154–1155. Report of failures of A.R.A. standard (D) couplers. Report of committee before Am. Ry. Assn. See also Ry. Age, vol. 80, no. 32, June 12, 1926, pp. 1736–1738, 1 fig.

CAR DUMPERS

Grain Elevators. Dumpimg Plant for Grain Cars. Engineering, vol. 121, nos. 3151 and 3153, June 4 and 18, 1926, pp. 650–652 and 716–720, 47 figs., partly on supp. plates. Details of Metcalf dumping plant for No. 3 grain elevator of Harbor Commissioners of Montreal; it comprises 4 dumpers, capable of unloading 7 cars per hr. containing 2500 bu. of wheat each.

CAR LIGHTING

Equipment and Maintenance. Requirements for Present Day Car Lighting. Ry. Elec. Engr., vol. 17, no. 7, July 1926, pp. 213–216, 1 fig. Discusses various types of equipment and quality of maintenance, which are vital factors for satisfactory illumination.

CAR WHEELS

Specifications. Report of Committee on Wheels. Ry. Age, vol. 80, no. 34, June 15, 1926, pp. 1830–1839, 12 figs. Developments in wrought-steel and cast-iron wheels; grinding of wheels; vertical flange defect; brake-burned wheels; etc. See also Ry. Rev., vol. 78, no. 25, June 19, 1926, pp. 1139–1142, 2 figs. Report before Am. Ry. Assn.

CARS, FREIGHT

Loading Rules. Committee on Loading Rules. Ry. Age, vol. 80, no. 30, June 11, 1926, pp. 1620–1625, 10 figs. Report of committee to Am. Ry. Assn.

Repairs. Progressive Car Repairs at Little Rock. Ry. Mech. Engr., vol. 100, no. 7, July 1926, pp. 439–440, 5 figs. Methods employed at Missouri Pacific shops; delivery of materials to fixed stations and moving cars from station to station effect marked reduction in cost.

Repairing Steel Cars. Ry. Jl., vol. 32, no. 7, July 1926, pp. 27–32, 15 figs. Describes unit spot system of repairing steel cars in railway shops; system affords systematic program of work and more efficient and economic operation of forces.

Specifications. Report of Committee on Car Construction. Ry. Age, vol. 80, no. 32, June 12, 1926, pp. 1713–1723, 13 figs. Sub-committee on fundamental calculations for design of box cars; test of cast-steel truck side frames; design of box cars; moisture content of lumber used in box cars; self-clearing hopper car; amplification of definitions designating letters for car equipment; lettering and marking cars, etc.

CARS, PASSENGER

Steel. Building All-Steel Railway Carriages. Machy. (Lond.), vol. 28, nos. 716 and 717, June 30 and July 8, 1926, pp. 349–355, 17 figs., and 381–387, 13 figs. June 30: Developments in steel passenger and freight cars; constructional principle of steel cars; assembly; details of jig system; direct-assembly methods. July 8: Operations on underframe members; sole-bar milling operation; notching, cutting, and drilling sole bars; producing underframe knees.

CARS, REFRIGERATOR

Bunkers. Divided Basket Bunkers Save Ice. Ry. Age, vol. 80, no. 38, June 26, 1926, pp. 1956–1959, 7 figs. Discusses economies to be obtained by use of wire basket bunkers divided by vertical air spaces which effect reduction in ice-carrying capacity and considerable increase in ice surface exposed to air circulating through bunker; results of test trips.

Iceless. North Shore Runs Iceless Refrigerator Cars. Ry. Rev., vol. 78, no. 24, June 12, 1926, pp. 1081–1083, 7 figs. These cars entirely eliminate necessity for precooling, icing service and salt drippings along right-of-way; refrigerating machine consists of 2½-ton-capacity Phoenix unit-type twin-cyinder ammonia compressor, driven by electric motor through Morse silent chain; describes process of refrigeration.

CARS, TANK

Specifications. Report of Committee on Tank Cars. Ry. Age, vol. 80, no. 32, June 12, 1926, pp. 1724–1732, 2 figs. Repairs to tanks; multiple-unit cars; tank with lock-bar joints; specifications; safety valves, dome covers and bottom outlet valves; patching of tank cars. Report before Am. Ry. Assn.

CASE-HARDENING

Irregular Carburization. Irregular Carburization of Iron and Iron Alloys—The Cause and Prevention, W. J. Merten. Am. Soc. for Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 907–920 and (discussion) 920–928 and 1004, 17 figs. Sets forth physico-chemical reactions within carburizing container; shows that chemistry of steel plays only minor role in success of satisfactory production of proper depth and uniformity of case and that physical structure of steel and mechanical arrangement of steel articles and carburizers for correct chemical reactions are major factors.

Nitride Process. Case Hardening with Nitrides. Iron Age, vol. 118, no. 6, Aug. 5, 1926, pp. 340–341, 5 figs. New process, not widely known, developed by Krupp; low temperatures and no quenching or drawing; deformation slight; suited to special steels.

Recuperation in. Recuperation in Carbonization of Steel, P. W. Hay. Am. Gas Jl., vol. 124, no. 24, June 12, 1926, pp. 515–517, 1 fig. Equipment of Buda Co. of Harvey, Ill., making use of gas-fired carbonizing furnace and recuperators; results of tests.

CAST IRON

Chilled. Chilled Iron Resembles White Iron, R. D. Billinger. Iron Age, vol. 118, no. 6, Aug. 5, 1926, pp. 345–346, 6 figs. Author seeks to show by means of photomicrographic study, similarity between chilled gray iron and typical white iron; type of castings under consideration is small piston rings of approximately ¼ in. cross-section.

Cooling Tests. Cooling Tests on Different Cast-Iron Test Pieces (Abkühlungsversuche an verschiedenen Gusseisenproben), F. Roll. Giesserei-Zeitung, vol. 23, no. 11, June 1, 1926, pp. 295–297, 7 figs. By means of cooling-off curves, temperature variations in cross-sections of different gray-iron castings are measured; observations were also made of structure and segregation; separate parts of test pieces form heat reservoirs that give off heat slowly to more rapidly cooling thinner cross-sections.

CENTRIFUGES

Design and Operation. Separation and Purification of Liquids by Means of Centrifugal Force (Zerlegung und Reinigung von Flussigkeiten durch Schleuderkraft), C. Schmitz. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 23, June 5, 1926, pp. 757–762, 14 figs. Design, operation and application of centrifuges with high speed and small drums for milk, oil, and other liquids.

Corrosion of. "Simply Cast Iron," H. I. Young. Indus. Chemist, vol. 2, no. 17, June 1926, pp. 254–257. Cast iron, in order to withstand corrosion conditions, must have certain composition and all its impurities must be controlled; discussion of problem with suggestions for elimination of difficulties.

Graphitization. Graphitization at Constant Temperature, H. A. Schwartz. Am. Soc. for Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 883–906, 5 figs. Mathematical analysis of data of graphitization; demonstrates that rate of graphitization is determined by rate at which carbon can migrate in iron; progress of graphite formation with time is shown to be expression of changing migratory distances, and concentration gradients produced by reaction; reference is made to range through which graphitizing rate varies in commercial material; effect of silicon on this constant. Bibliography.

High-Grade. High-Grade Cast Iron (Ueber hochwertigen Grauguss), T. Klingenstein. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 12, Mar. 20, 1926, pp. 387–390, 15 figs. Production of special castings with special consideration of the Wüst furnace.

Pearlitic. A Note on the Properties of "Perlit" Iron, A. E. M. Smith. Foundry Trade Jl., vol. 33, no. 514, July 1, 1926, pp. 13–16, 8 figs. Gives characteristics of perlit iron and describes Lanz process for manufacturing perlit-iron castings.

Quality. Results Obtained in the Improvement of the Qualities of Cast Iron, L. Piedboeuf. Foundry Trade Jl., vol. 33, no. 514, June 24, 1926, pp. 496–499, 5 figs. Reviews research work on improvement of cast-iron quality, discussing particularly Maurer's cast-iron diagram.

Test Bars. Test Bars for Gray Cast Iron (Note sur une éprouvette de coulée pour la fonte grise), P. Corbellini. Fonderie Moderne, no. 20, May 1926, pp. 109–111, 6 figs. Proposes use of test pieces as in Keep's test and shows advantages by being able to tell content of silicon from depth of pipe, etc.

CENTRAL STATIONS

Design. Power Station Design, V. E. Alden. Elec. Light & Power, vol. 4, no. 7, July 1926, pp. 17–18, 84 and 86, 2 figs. Before designing new power station, ideals or objectives of station should be clearly defined; basic facts as to conditions affecting construction should be determined and accurate forecast should be made of conditions under which station will operate.

England. Power at Port Sunlight. Power Engr., vol. 21, no. 243, June 1926, pp. 209–219, 18 figs. Principal features of power-supply system of Lever Bros. at their group of factories in Cheshire.

Glasgow. Glasgow Corporation Electricity Supply. Elec. Rev., vol. 98, no. 2534, June 18, 1926, pp. 899–902, 7 figs. Developments of Dalmarnock generating station; station was designed for ultimate capacity of 200,000 kw.; generator and outgoing feeder switchgear is of metal-boxed, oil-immersed type; each of 6500-volt alternators has its own bank of 3 single-phase transformers; equipped with eight boilers of Babcock marine type.

Lignite-Burning. Progress in Lignite Power Plant Construction in Germany, A. Peucker. Elec. World, vol. 87, no. 26, June 26, 1926, pp. 1392–1393. Changes in methods of fuel and ash handling, additions to boiler plant and turbine installation and raising of power factor from 0.67 to 0.90 greatly increase capacity of Zschornewitz power house.

Ohio State University. New Plant Supplies Power, Light and Heat to Ohio State University. Power Plant Eng., vol. 30, no. 13, July 1, 1926, pp. 728–734, 14 figs. Three condensing turbo-generators, one extraction, hot-water heating system for buildings; 2-phase power system and unique arrangement of fan ducts are among features of interest in power plant at Columbus, O.

Sterlington, La. Heat Balance at the Sterlington Power Station, W. S. Johnston. Power Plant Eng. vol. 30, no. 13, July 1, 1926, pp. 735–739, 6 figs. Extraction feedwater heaters, high-pressure evaporators with high heat-level condensers and house turbine form scheme of station of Louisiana Power Co. near Monroe, La.

Toledo, O. Toledo Edison Increases Capacity of Acme Plant, E. A. Doerr. Power Plant Eng., vol. 30, no. 12, June 15, 1926, pp. 678–683, 6 figs. Addition of four 18,000-sq. ft. boilers and auxiliaries, together with stage heaters to improve heat balance, enables turbines to operate at full load.

CHIMNEYS

Calculation. Calculation of Chimneys for Central-Heating Boilers (Ueber Schornsteinberechnung für Central-Heizungskessel), Seelig. Verband der Centralheizungs-Industrie Mitteilungen der Wärmetechnischen Abteilung, Jan.-Mar. 1926, pp. 32–38. Calculation of chimney before it is built, so that it can be proportioned to dimensions of boiler, and in conformity with Berlin building code.

COAL

Carbonization. Low-Temperature Carbonization of Fuel, D. Brownlie. Power House, vol. 19, no. 13, July 5, 1926, pp. 19–20, 42 and 44, 3 figs. Different processes of carbonization, operated in conjunction with steam boiler, general furnace, and power plant.

Direct Conversion to Electric Energy. Can the Heat of Combustion of Coal Be Turned Directly into Electric Energy? J. G. A. Rhodin, Engineer, vol. 142, no. 3680, July 23, 1926, pp. 80–81. Estimates approximately what can be got out of ton of coal by first removing volatile matter by high- or low-temperature carbonization; points out intrinsic difficulties of finding new way to generate power.

Heating-Value Determination. Graphical Method of Determining the Heating Value of Coal. (Méthode graphique de détermination du pouvoir calorifique du charbon), A. de Zawaritxky. Chaleur & Industrie, vol. 7, no. 72, Apr. 1926, pp. 191–199, 10 figs. Ten charts and instructions for their use.

Pulverized. See PULVERIZED COAL.

COAL HANDLING

Bridges. 20-Ton Electric Coaling Transporter. Engineering, vol. 122, no. 3155, July 2, 1926, pp. 5–7, 2 figs. Built for Queensland Government for rapid loading of coal into vessels from pier 500 ft. long and 84½ ft. wide.

Conveyors. Conveyors for Coal and Ash Handling in Boiler Plants (Der gegenwärtige Stand des Transportwesens zur Bekohlung und Entaschung von Dampfkesseln), F. Riedig. Fördertechnik u. Frachtverkehr, vol. 19, nos. 11 and 12, May 28 and June 11, 1926, pp. 157–160 and 176–180, 25 figs. Discusses systems of mechanical handling of coal, types of bucket conveyors, conveying bands, spirals and tubes; also transportable types; cableways, ash handling by suction and hydraulic system, combination of mechanical and wet process.

Machinery. Coal-Handling Machinery at Hull. Shipbldg. & Shipg. Rec., vol. 27, no. 23, June 10, 1926, pp. 612–614, 4 figs. 380-ft. electrically driven belt conveyors and hydraulic wagon-tipping machinery at Victoria Dock of London & North Eastern Ry.

Unloaders. Hydraulic Unloading of Coal (Le déchargement hydraulique du charbon), P. Calfas. Génie Civil, vol. 88, no. 13, Mar. 27, 1926, pp. 289–292, 6 figs. Plant of St. Ouen central station, where coal is unloaded by means of strong water jets, cars being run out crosswise on inclined track, coal and water running into canal, along track, and water being re-used indefinitely; used for rapid unloading of large quantities of coal.

COLD STORAGE

Temperature Measurement. The Measurement of Temperature and Carbon Dioxide in Cold Stores. Ice & Cold Storage, vol. 29, no. 340, July 1926, pp. 181–184, 9 figs. Describes thermometers of electrical-distance type used to observe from one convenient position all temperatures throughout cold-storage room.

CONNECTING RODS

Machining. Rods for the Hupp "Six," F. H. Colvin. Am. Mach. vol. 65, no. 1, July 1, 1926, pp. 9–10, 5 figs. Holding fixtures for rods; elongating holes in large ends; milling bolt bosses and cutting off caps; oil-grooving device.

CONVEYORS

Belt. Material Handling with Belt Conveyors. Buildings, vol. 65, no. 6, June 23, 1926, pp. 261–263, 6 figs. How aggregate for 9-story reinforced-concrete building at Atlanta, Ga., was conveyed 400 ft.

Screw. The Screw Conveyor and Its Manufacture, D. C. Wright. Am. Mach., vol. 64, nos. 24 and 25, June 17 and 24, 1926, pp. 931–933 and 977–980, and vol. 65, no. 1, July 1, 1926, 19 figs. Different types of screw conveyors; design of sectional flight conveyor; efficiency in manufacture; methods of production for flight sections. June 24: Methods of piercing rivet holes in sectional flight circles; forming or shaping sections; continuous flight conveyors and their construction. July 1: Manufacturing conveyor backbone; shrinking collars and pipes; assembling sectional and continuous flight conveyors.

COOLING TOWERS

Reinforced-Concrete. Ferro-Concrete Cooling Towers. Can. Engr., vol. 50, no. 24, June 15, 1926, p. 665. Large circular structures advocated, with inlet piping in center; concrete is rigid and light.

COST ACCOUNTING

Systems. Estimating the Cost of a New Product. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, pp. 243–246, 5 figs. Describes possibilities of scientific costing system and indicates far-reaching character of advantage to be derived by making full use of statistics thus obtained.

COTTON MILLS

Fundamental Measurements. Fundamental Measurements in a Cotton Mill. Mech. Eng., vol. 48, no. 7, July 1926, pp. 746–748. Discussion of paper by S. S. Paine presented at Providence meeting of Am. Soc. Mech. Engrs.

CRANES

Gasoline-Electric. Petrol-Electric Mobile Crane. Ry. Gaz., vol. 44, no. 25, June 18, 1926, pp. 785–787, 5 figs. Crane, which has been supplied to several railway companies in England and abroad, is adaptable to variety of purposes.

Storm Brakes. Storm Brakes for Cranes and Transporters. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, pp. 239–240, 4 figs. Describes two mechanical devices by means of which cranes and loading and unloading bridges and telphers are automatically secured to their rail tracks to prevent damage by storm; appliances, which are of German origin, have brake power capable of resisting pressure of 20 tons.

Truck. Raising and Turning Superstructures by Means of Crane Trucks (Ausheben und Verschwenken von Ueberbauten mit Kranwagen), O. Miehlke. Bautechnik, vol. 4, no. 24, June 4, 1926, pp. 337–344, 13 figs. Discusses raising and exchanging of small bridges, etc., by truck cranes at either side; calculation of power; simultaneous management of cranes; difficulties in operation.

CRANKSHAFTS

Diesel-Engine. Forging and Twisting a Three-Throw Crankshaft for Diesel Engines (Das Schmieden und Verdrehen von dreifach gekröpften Kurbelwellen für Dieselmaschinen), P. Schweissguth. Werkstattstechnik, vol. 20, no. 10, May 15, 1926, pp. 306–307, 17 figs. Older and more recent methods of forging; elongation of fiber in twisting; precautions in twisting and subsequent precautions.

CUPOLAS

Reactions, Chemistry of. Some Notes on the Chemistry of the Cupola, F. C. Thompson. Foundry Trade Jl., vol. 33, no. 514, June 24, 1926, pp. 491–495, 4 figs. General discussion of chemistry of reactions involved in cupola practice; thermal balance of cupola; reactions with carbon; lime or limestone; reactivity of coke and addition of further air through auxiliary tuyeres; bibliography.

CUTTING METALS

Turning. Temperature of Cutting Point in Lathe Work and Dependence on Form of Chip Cross-Section (Die Schneidentemperatur beim Drehen in Abhängigkeit von der Form des Spanquerschnittes), K. Gottwein. Maschinenbau, vol. 5, no. 11, June 3, 1926, pp. 505–506, 3 figs. At given values of chip cross-section and cutting-speed, temperature at point of tool is affected by ratio of depth of cut to feed; temperature drops if this ratio increases; gives number of micrographs in systematic order for comparison.

CYLINDERS

Automobile, Machining. Fixtures and Methods in Machining Hupp Cylinders, F. H. Colvin. Am. Mach., vol. 64, no. 25, June 24, 1926, pp. 985–988, 12 figs. Illustrations of adaptation of existing equipment; conveyors; speed operations; system of tagging shows what operations have been completed.

D

DIES

Brake Drums. Dies for Making Brake Drums. C. S. Wheeler. Iron Age, vol. 117, no. 25, June 24, 1926, pp. 1774–1775, 2 figs. Describes two dies employed in manufacture of brake drums, one being combination blanking and drawing die, and other piercing die; method of figuring dimensions.

DIESEL ENGINES

Electric Drive vs. Comparative Calculations of the Cost of Power from Diesel Engines and from High-Tension 3-Phase Current (Vergleichsberechnung zwischen Antrieb durch Dieselmotor und Bezug von hochgespanntem Drehstrom), Hack. Chemiker-Zeitung, vol. 50, no. 48, Apr. 28, p. 301. Calculation of direct and indirect operating costs for Diesel engines and for electric motors.

15,000-Hp. The 15,000-Hp. Diesel Engine, M.A.N. Type, Built by Blohm & Voss for Hamburg Central Station (Der 15,000 PS-Dieselmotor, Bauart MAN, erbaut von Blohm & Voss für die Hamburgischen Elektricitätswerke), W. Ladahn. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 24, June 12, 1926, pp. 818–825, 18 figs. Details of double-acting 9-cylinder Diesel engine driving 13,000-kva. generator at plant in Neuhof; consisting of 3 parts, 9-throw crankshaft is forged of Siemens-Martin steel. See also description in Oil Engine Power, vol. 4, no. 7, July 1926, pp. 416–420 and 430, 8 figs.

Fuel-Supply System. Fuel Supply System for the Diesel Engine, C. C. Hermann. Power Plant Engr., vol. 30, no. 13, July 1, 1926, pp. 744–745, 1 fig. Care needed in estimating size of storage tank required as well as in its installation and piping.

Generator Set. A 10,000 KW. Diesel-Generator Set. Engineer, vol. 141, no. 3676, June 25, 1926, pp. 662–663, 5 figs. Details of 15,000-b.hp. engine and 10,000-kw. 3-phase generator installed in Neuhof power station, Hamburg; engine is coupled direct to alternator which is believed to be largest of kind ever built for Diesel-engine work.

High-speed. A French High-Speed Diesel, R. H. Andrews. Power, vol. 64, no. 5, Aug. 3, 1926, pp. 162–163, 2 figs. Details of Peugeot-Tartrais engine which is lightest oil engine built, weighing only 11 lb. per hp.; it is first to undergo successful road test in automobile drive.

Lubrication. Large Diesel Engines and Their Lubrication. Lubrication, vol. 12, no. 6, June 1926, pp. 61–72, 14 figs. Discusses constructional features with two and four-cycle principles; methods of fuel injection and fuel efficiency; essential characteristics of lubricants and air-compressor lubrication.

Marine. Marine Diesel Machinery Arrangement, C. C. Pounder. Mech. World, vol. 79, nos. 2046, 2050, 2054, and 2058, Mar. 19, Apr. 16, May 14 and June 11, pp. 223–224, 302–303, 378–379, and 447–449, 9 figs. Salient points involved in arrangement of Diesel engines on board ship.

Spray Atomization vs. Ante-Chamber Process. Spray Atomization vs. Ante-Chamber Process (Druckeinsprutzung oder Vorkammerverfahren), F. Modersohn. Zeit. des Vereines deutscher Ingenieure, vol. 70, nos. 23 and 26, June 5 and 26, pp. 767–771 and 891–894, 23 figs. Spray-atomizing engines; control of piston heat, open and closed nozzles. Ante-chamber engines: development, phenomena in ante-chamber, fuel injection, cooling, starting. Relative merits of spray atomizing and ante-chamber; disadvantages of both processes; conclusions.

Torsional Vibration. Torsional Vibration in the Diesel Engine, F. M. Lewis. Marine Eng. & Shipping Age, vol. 31, no. 7, July 1926, pp. 415–423, 5 figs. Amplitude of vibration at critical speeds in multi-cylinder engines; damping forces; elastic hysteresis; elimination of vibration; mathematical analysis.

Worthington. The Worthington Double-Acting Two-Cycle Diesel Engine. Diesel & Oil Engine Jl., vol. 2, no. 4, Apr.-May-June 1926, pp. 14–22, 14 figs. Discusses advantages of engine and compares it with other types.

DISKS

Rotating, Stresses in. The Effect of Axial Restraint on the Stress in a Rotating Disk, W. G. Green. Lond., Edinburgh & Dublin Philosophical Mag. & Jl. of Sci., vol. 1, no. 6, June 1926, pp. 1236–1251, 3 figs. Study of problem of rotating disk carried on shaft; presence of shaft prevents parts of disk near axis shifting so far in direction of axis as they would if there were no shaft; illustrative problem is given to trace consequences of this effect.

DRAWINGS

Drafting-Room System. A Simple Drafting-Room System, E. H. Bruce. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 957–961, 15 figs. In this system author has endeavored to eliminate, as far as possible, various factors that cause delays in shop and drafting room.

DRILLING

Deep-Hole. Deep-Hole Drilling, F. W. Bach. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 910–911, 2 figs. Results obtained in drilling deep holes in barrels of shotguns and rifles.

DRILLING MACHINES

Plate and Rivet-Hole. Cincinnati Bickford Plate and Rivet-Hole Driller, Heavy Type. Am. Mach., vol. 65, no. 5, July 29, 1926, pp. 219–220, 2 figs. Designed for boiler, tank and structural shops. See also Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 993–

International Exhibition, England. The International Foundry Trades Exhibition. Engineering, vol. 121, nos. 3152, 3153, and 3154, June 11, 18, and 25, 1926, pp. 688–691, 720–723, and 753–755, 26 figs., partly on p. 728 and supp. plates. Review of exhibition at Royal Agricultural Hall. See also Engineer, vol. 141, no. 3675, June 18, 1926, pp. 630–633, 9 figs.

Iron-Mixing Furnaces. Mixing Blast Furnace and Cupola Iron, H. Illies and A. Hesse-Wortmann. Iron Age, vol. 118, no. 6, Aug. 5, 1926, p. 350, 1 fig. German practice in making pipe employs large mixer or furnace heated with oil; life of apparatus a feature.

Quantity Production. Foundry Quality in Quantities, A. Lenz. Can. Foundryman, vol. 17, no. 6, June 1926, pp. 7–15, 13 figs. Factors effecting quantity production in foundries; control of mills; elimination of risers; need of close checking; floor and conveyor molding; assembling of cores; keeping stock at minimum, etc. See also Foundry Trade Jl., vol. 33, no. 514, June 24, 1926, pp. 481–485, 7 figs.

Study Courses. Course in Foundry Studies. Iron Age, vol. 118, no. 5, July 29, 1926, pp. 279–281, 2 figs. Indianapolis High School prepares students to enter foundries; short course for general student.

FOUNDRY EQUIPMENT

International Exhibition, London. International Foundry and Allied Trades' Exhibition. Foundry Trade Jl., vol. 33, no. 512, June 10, 1926, pp. 413–428, 21 figs. Description of exhibits.

Molding Boxes. The Making of "Sterling" Boxes. Foundry Trade Jl., vol. 33, no. 512, June 10, 1926, pp. 406–407, 4 figs. Methods and equipment in manufacture of rolled steel foundry molding boxes at works of Sterling Foundry Specialties, Bedford, Eng.

Production-Increasing. Costs Show Lower Trend on Installing New Foundry Equipment, P. Dwyer. Foundry, vol. 54, no. 12, June 15, 1926, pp. 464–468 and 478, 7 figs. Details of equipment which has directly or indirectly increased production from 200 to 600 tons per month; ladle-handling devices; sand-handling equipment.

Sand-Handling. Materials-Handling Equipment in Foundries (Fördervorrichtungen in Giessereien), Walther. Glasers Annalen, vol. 98, no. 10, May 15, 1926, pp. 163–166, 6 figs. Describes equipment of Eickhoff Machine Works foundry at Bochum, Germany, especially for handling of sand, main principle of which is to keep labor in place and move material whenever possible.

FREIGHT HANDLING

Rail and Highway Transportation. Co-Ordination of Rail and Highway Freight Transportation Within the Terminal Area, F. J. Scarr. West. Soc. Engrs.—Jl., vol. 31, no. 4, Apr. 1926, pp. 128–134. Shows that highway transportation has definite field; concludes that rail transportation, poorly adapted to efficiently handle terminal transportation, will gradually be relieved of responsibility for performing so-called retail service and be permitted to confine its efforts to that for which it is best suited, handling of mass transportation over long distances; highway transportation will ultimately assume major role in intra-terminal complete movement of freight; inter-line interchange of l.c.l. freight, and movement of freight between rail carrier and trader's door.

Terminal Transfer. Saving 100,000 Car Days a Year, G. E. Boyd. Ry. Rev., vol. 78, no. 26, June 26, 1926, pp. 1193–1198, 5 figs. Columbia Terminal Co. of St. Louis handles nearly all l.c.l. freight interchanged between railroads in St. Louis terminal area; use system of carting in which tractors and semi-trailers are employed; economy is emphasized.

Fuels. See COAL; OIL FUEL; PULVERIZED COAL.

FURNACES, HEATING

Study of. A Study of Heating Furnaces. (Étude sur les Fours poussants), M. Laffargue. Chaleur & Industrie, vol. 7, no. 73, May 1926, pp. 264–268, 2 figs. Necessity for short flame; working temperatures; fuel requirements per ton of metal heated; length of tunnel kiln; calculation of furnace.

G

GAGES

Caliper Thread. Thread Caliper Gauges, G. A. Tomlinson. Machy. (London), vol. 28, no. 715, June 24, 1926, pp. 323–325, 2 figs. Discusses particularly Wickman type; explains conditions to be satisfied to avoid interference between gage and helix of screw; importance of correct alignment of two anvils; method of measuring this type of gage, recently developed.

Dial-Reading. Krupp Micro-Contact Gages (Kruppsche Mikrotaft-Geräte), M. Bartholdy. Kruppsche Monatshefte, vol. 7, Apr. 1926, pp. 62–69, 10 figs. Dial-reading snap gages for checking circularity and taper, both external and internal.

Limit Specifications. Accuracy Control in Interchangeable Manufacture. Machy. (Lond.), vol. 28, no. 716, June 30, 1926, pp. 361–364, 6 figs. System for specifying limits on parts and gages from standardized lists; method of gage inspection.

National Standards. National Standards for Fits and Gages, J. Gaillard. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 927–932, 9 figs. Comparison of standards established in United States, Great Britain, Holland, Sweden, Germany, Austria, and Switzerland.

Tapers, for. Accurate Gaging of Tapers, A. J. Cayouette. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 945–947, 7 figs. Gages used for checking taper parts made to basic diameter on which no tolerance is specified.

GAS ENGINES

Indicators. Indicators for Gas Engines. Gas Engr., vol. 42, no. 602, June 1926, pp. 125–127, 9 figs. Various designs of indicators recently developed and precautions necessary in their use.

GAS PRODUCERS

Mechanical. Mechanical Gas Producers, J. F. Rogers and V. Windett. Mech. Eng., vol. 48, no. 7, July 1926, pp. 743–746, 5 figs. Problems arising in substitution for hand-operated producers of completely mechanical producer in which manual labor is supplanted by mechanical operations.

GASES

Radiation of Combustion. Gas Radiation in Connection with Flow in Pipes (Die Gasstrahlung bei der Stremung im Rohr), W. Nusselt. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 23, June 5, 1926, pp. 763–765, 2 figs. Formulas for calculation of gas radiation of a cylindrical gas column; refers to author's previous work on heat transmission in internal-combustion engines and states that values and formulas developed therein for gas radiation can be readily applied to calculation of radiation of hot combustion gases in furnaces and steam boilers.

GEAR CUTTING

Shaper Cutters. Design and Application of Gear-shaper Cutters, H. Walker. Machy. (Lond.), vol. 28, no. 711, May 27, 1926, pp. 199–201, 5 figs. Cutting internal gears.

GEARS

Hobbed. The Generation of Tooth Forms by a Hob, E. A. Limming. Machy. (Lond.), vol. 28, no. 717, July 8, 1926, pp. 397–399, 3 figs. Analysis of action of hob when generating tooth forms for spur or helical gears.

Measuring Instrument. Gear Measuring Equipment. Machy. (Lond.), vol. 28, no. 714, June 17, 1926, pp. 301–304, 9 figs. Gear-measuring instrument made by Maag Co., Zurich, Switzerland; measurements obtained with this instrument are made with reference to theoretically correct involute curves; instrument for examining involute tooth profiles and comparing these with correct theoretical shape; also serves to examine symmetry of two profiles of tooth.

Mounting on Structural Work. Mounting Gearing on Structural Work, E. G. Fiegehen. Mech. World, vol. 79, no. 2057, June 4, 1926, pp. 426–427, 7 figs. Summary of points to be observed in mounting gearing upon structural work.

Planetary. New French Planetary Gear Controlled by Magnetic Clutches. Automotive Industries, vol. 54, no. 24, June 17, 1926, pp. 1045–1046, 3 figs. Transmission invented by M. Cotal is of all-spur type; silent in operation and eliminates physical effort for gear shifting; trials made by Voisin.

Reversals of Circular Motion, F. D. Furman. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 875–879, 10 figs. Also Machy. (Lond.), vol. 28, no. 717, July 8, 1926, pp. 392–396, 9 figs. Planetary gear mechanism which provides reversals without aid of shifting device; graphical analysis and analytical solution of reversing mechanism.

Teeth, Standardization of. Standardization of Gear Teeth in Germany. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 938–939, Information based upon report by Bauersfeld, chairman of Committee on Gear Teeth of N.D.I. (Normenausschuss der Deutschen Industrie); basic rack and pressure angle; total tooth depth; tooth thickness and clearance; radius of fillet at root of tooth—modification of profile.

Tooth Forms. Spur Gear Tooth Form. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 905–906, 1 fig. Reasons underlying tentative American standard which has been completed by Sub-committee No. 4 of Sectional Committee on Standardization of Gears, and approved by Am. Eng. Standards Committee; problem related to standardization of gear-tooth forms.

The Design of Gear Tooth Forms, E. Buckingham. Am. Mach., vol. 64, nos. 8, 9, 10, 11, 12, 13, 14, 15, 16, 18, 19, 21, 23, 24, and 25, Feb. 25, Mar. 4, 11, 18, 25, Apr. 1, 8, 15, 22, May 6, 13, 27, June 10, 17, and 24, pp. 299–302, 345–347, 391–394, 439–441, 465–467, 509–512, 555–562, 595–598, 635–637, 719–722, 745–751, 817–820, 895–898, 945–948, and 973–975, 69 figs. Attempt to solve present-day gear problems by employing fundamental principles as foundation. Feb. 25: Discusses conjugate tooth action, involute curve, involute trigonometry, involute tooth design, strength of gear teeth and their durability under load. Mar. 4: Construction of line of action from given tooth profile and reverse; requirements for interchangeable tooth forms; cycloid curve. Mar. 11: Characteristics of cycloid, epicycloid and hypocycloid curves; application to tooth forms and resulting lines of action; cycloidal blower rotors. Mar. 18: Basic racks to generate spline shafts; theory of involute curve and its chief characteristics when used as gear tooth profile form. Mar. 25: Action of involute against straight line; summary of involute curve properties; its application to gear tooth profiles; duration of contact. Apr. 1: Combined rolling and sliding takes place between 2 involutes in contact; calculation of ratio of sliding. Apr. 8: Tooth and bearing pressures; simple solution of involute tooth gear problems through use of involute functions in tabulated form. Apr. 15: Further solutions of general involute tooth-gear problems; relation of gear to meshing rack; duration of contact; undercutting. Apr. 22: Nomenclature and definitions of gear teeth; 14½-deg. composite of gear teeth with tabulated formulas for their proportions; May 6: Form milling of gears and action of gears produced by this method; effect of eccentricity; effect of errors in Milling. May 13: Tabular data on correcting tooth depth with various combinations of range milling cutter. May 27: Introduction of straightsided hobs and racks resulted in 14½ deg. generated gear tooth system; effect of undercutting on length of contact. June 10: 20-deg., full-depth tooth gear system; tooth proportions and length of contact; June 17: 20-deg. stub-tooth gear system for gears of small number of teeth. June 24: 14½-deg. variable-center system.

The Design of Gear Tooth Forms, E. Buckingham. Am. Mach., vol. 65, no. 5, July 29, 1926, pp. 195–198, 3 figs. Maag gears; gear trains; study of modified pinions of small numbers of teeth of proportional center distance system; contact and sliding conditions.

GOVERNORS

Constant- and Variable-Speed. Constant-Speed and Variable-Speed Governors. Engineering, vol. 122, no. 3157, July 16, 1926, pp. 87–88, 4 figs. Governor designed primarily for controlling speed of small and medium size internal-combustion engine and small steam turbines running at high speed, also governor in which arrangements are such that load on spring is constant throughout whole speed range; both types are designed by F. O. L. Chorlton.

GRINDING

Automobile Parts. Grinding Franklin Car Parts, F. B. Jacobs. Abrasive Industry, vol. 7, no. 7, July 1926, pp. 206–207, 5 figs. Centerless grinding machines are employed for number of operations; other diameter of brake drums ground; cylinders finished by honing.

Railway Shops. German State Railway Shops Employ Precision Grinding, H. Hermanns. Abrasive Industry, vol. 7, nos. 5, 6 and 7, May, June and July 1926, pp. 145–148, 187–188 and 221–223, 14 figs. Describes number of interesting grinding operations performed in repair shops of German State railways.

Screw Threads. Grinding Small Internal Screw Threads, O. S. Marshall. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, p. 950, 2 figs. Describes method employed.

GRINDING MACHINES

Automobile Parts. High Production Drilling and Boring of Automobile Components. Brit. Machine Tool Eng., vol. 3, no. 39, May-June 1926, pp. 420–424, 5 figs. Special drilling and boring equipment for surface-grinding operation of automobile rear-axle frame.

Internal. An Ingenious Internal Grinder. Automobile Engr., vol. 16, no. 216, June 1926, pp. 224–225, 4 figs. Fully automatic machine with hydraulic table drive.

Locomotive Axles. Locomotive Axle Grinding. Brit. Machine Tool Eng., vol. 3, no. 39, May-June 1926, pp. 407–408, 2 figs. Describes 18-in. by 84-in. plain grinding machine equipped for grinding of locomotive axles built by Churchill Grinding Machine Co.

Roll. Two Interesting Roll Grinding Machines. Machy. (Lond.), vol. 28, no. 714, June 17, 1926, pp. 299–300, 3 figs. Model "D" plain grinding machine made by Churchill Machine Tool Co. fitted with special cambering motion; it is traveling-table type used for grinding of rolls requiring very high accuracy; also another machine of traveling-table type, but built specially for grinding rolls.

GUN METAL

Castings. Gun-Metal Castings. Automobile Engr., vol. 16, no. 216, June 1926, pp. 217–218, 1 fig. Notes on composition and production.

H

HAMMERS

Manufacture. Manufacture of the Kango Hammer. Machy. (Lond.), vol. 28, no. 713, June 10, 1926, pp. 267–271, 15 figs. Production methods employed by Coventry Gage & Tool Co.

Pneumatic. 30-cwt. Pneumatic Hammer. Machy. (Lond.), vol. 28, no. 715, June 24, 1926, p. 341, 3 figs. Hammer recently developed giving maximum horsepower for full blows of 75, requiring 50 hp. continuous rating in motor and having stroke of 39-in. clear space between slides.

HARDNESS

Rockwell and Brinell. Some Comparisons Between Rockwell and Brinell Hardness, R. C. Brumfield. Am. Soc. Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 841–856, 7 figs. Results obtained in testing both hard and soft metals by Rockwell and Brinell hardness tests; establishes relation between Brinell hardness and each type of Rockwell hardness in which most influential factor is area of contact with piece tested; tables show comparative Rockwell and Brinell hardness for various metals.

Testers. The Durometer—An Instrument for Testing Hardness, A. Suaveur. Am. Soc. for Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 929–932 and 1000, 2 figs. New instrument used in determining hardness of metals and manner in which it functions.

The Firth Hardometer. Mech. World, vol. 80, no. 2061, July 2, 1926, pp. 3–4, 5 figs. Describes hardometer hardness-testing machine developed by Thomas Firth & Sons, Ltd., Sheffield, England, designed to fulfill all workshop requirements at remarkably reasonable initial cost.

HEAT TRANSMISSION

Calculation. Heat Transmission from Solids to Fluids in Turbulent Flow (De warmteovergang van vaste lichamen op turbulent stroomende vloeistoffen),

994, 2 figs. See also Iron Age, vol. 118, no. 5, July 29, 1926, pp. 282–283, 2 figs.

Multiple-Spindle. A Study of Drilling Machines, J. G. Horner. Machy. (Lond.), vol. 28, nos. 707 and 715, Apr. 15 and June 24, 1926, pp. 73–77, 12 figs., and 331–337, 15 figs. Apr. 15: Multiple-spindle drills of gang type; cluster system. June 24: Pneumatic and electric drilling machines.

Rapid Drilling and Tapping on Multiple-Spindle Machines, O. R. Reller. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 860–862, 6 figs. Drilling connecting rods, washing-machine parts, holes at angles; tapping flanges on multiple-spindle machines.

Small Multiple-Spindle Drilling Machines. Machy. (Lond.), vol. 28, no. 713, June 10, 1926, pp. 279–280, 4 figs. Made by G. Boley, Esslingen; has 13 spindles driven from gear box at head; spindles run in phosphor-bronze bushes and can be removed in few moments.

DURALUMIN

Heat Treatment. Effect of Heat Treatment on Duralumin, R. J. Anderson. Forging—Stamping—Heat Treating, vol. 12, nos. 5 and 6, May and June 1926, pp. 169–172 and 208–211 and 219, 32 figs. Summary of results of study on effect on various heat treatments on microstructure of duralumin sheet; object of work was to examine effects of wide variety of heat treatments and throw light on age-hardening phenomenon. June: Effect of temperature of quenching and aging; effect of time period of aging; effect of quenching medium; features of microstructure. (Abstract.) Thesis submitted to Mass. Inst. Technology.

E

ECONOMIZERS

Heat Transmission in. Temperatures and Heat Exchange in a Cast-Iron Economizer. (Temperaturen und Wärmeaustausch an einen gusseisernen Speisewasservorwärmer), H. Reiher and G. Neidhardt. Archiv für Wärmewirtschaft, vol. 7, no. 6, June 1926, pp. 153–157, 3 figs. Gives results of measurements of temperatures and flue-gas velocity in a Green economizer, from which coefficient of heat transmission is calculated.

EJECTORS

Three-Stage Air. Three-Stage Air Ejector Uses Less Steam, D. W. R. Morgan. Power, vol. 64, no. 4, July 27, 1926, pp. 127–128, 2 figs. Westinghouse Co. has completed development of three-stage condensing-type ejector, consisting of three single-nozzle ejectors in series, with combined condenser and cooler interposed after each stage, third stage finally discharging into aftercooler.

ELECTRIC FURNACES

Carburizing. Progressive Carburization in Rotary Electric Furnaces, H. E. Martin. Am. Soc. for Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 933–937, 1 fig. Reasons why electrical heat was selected for additional carburizing and heat-treating furnaces at plant of Dodge Bros.; details of rotary carburizing furnaces, cycle of operations, heat-control arrangement and advantages.

Castings and Ingots. Electric Furnaces for Castings and Ingots, E. T. Moore. Elec. Jl., vol. 23, no. 6, June 1926, pp. 279–284, 9 figs. Details of Edmoore electric furnace of rolling-rocker type, installed by Reading Steel Casting Co.; it has capacity of 7 to 15 tons, is equipped with three electrodes, and circular shell composed of boiler plate having inside diameter of 12½ ft.

Heat-Treating. Electric Furnaces for Heat Treating, A. N. Otis. Forging—Stamping—Heat Treating, vol. 12, no. 6, June 1926, pp. 192–196, 9 figs. Typical installations in Schenectady plant of General Electric Co., and variety of heat-treating work to which they have been adapted.

Power Factor. Power Factor of Electric Furnaces (Le facteur de puissance des fours électriques), M. Bergeon. Société Française des Electriciens—Bul., vol. 6, no. 55, Mar. 1926, pp. 292–296, 3 figs. Establishes empirical formula, showing that it is desirable to reduce frequency and increase voltage and multiply circuits as was done by Miguet in his 100,000 A furnace.

ELECTRIC LOCOMOTIVES

Brazil. Powerful New Electric Locomotive for Brazil. Elec. Ry. & Tramway Jl., vol. 54, no. 1345, May 14, 1926, pp. 239–240, 2 figs. 100-ton locomotive built by Metropolitan-Vickers Electrical Co. for passenger service on Paulista Ry. of Brazil; new locomotive is most powerful yet constructed in Great Britain, having 6 motors totalling 2340 hp.

Characteristics. Report of Committee on Electric Rolling Stock. Ry. Age, vol. 80, no. 35, June 16, 1926, pp. 1853–1863, 8 figs. Tractive-power characteristics of electric locomotives; probable effect of height of center of gravity; locomotives of Virginian electrification; design of multiple-unit and trailer cars for terminal service of trunk-line railroads; electrification progress. See also Ry. Rev., vol. 78, no. 25, June 19, 1926, pp. 1142–1149, 4 figs. Report before Am. Ry. Assn.

Motor-Generator. Motor-Generator Locomotives for the Great Northern. Ry. Age, vol. 81, no. 1, July 3, 1926, p. 20. Two locomotives weighing 250 tons each, under construction for Great Northern railway; maximum tractive effort based on 30 per cent coefficient of adhesion is 122,940 lb.; total wheelbase, 60 ft.

Regenerative Braking. Regenerative Braking for irect-Current Locomotives, A. Bredenberg, Jr. Am. Inst. Elec. Engrs.—Jl., vol. 45, no. 7, July 1926, pp. 613–619, 8 figs. Comparative study of several d.c. regenerative-braking systems now in successful operation; general characteristics, including advantages, limitations and functions.

Single-Phase. A Single-Phase 50-Cycle Electric Locomotive. Engineer, vol. 141, no. 3674, June 11, 1926, pp. 606–608, 11 figs. Built by F. Krupp Corp., Essen, Germany; equipped with novel type of compensated single-phase motor without commutator, which is claimed to possess running qualities of 3-phase motor.

Swiss Federal Railways. New Electric Express Locomotives, Swiss Federal Railways. Ry. Gaz., vol. 44, no. 25, June 18, 1926, pp. 783–784, 2 figs. Design of 4-6-2 locomotives weighing 54 tons and having maximum speed of 56¼ miles per hour.

ELECTRIC RAILWAYS

Cars, Articulated. Hamburg Stadt- Und Vorortbahnen Articulated Coaches. Elec. Ry. & Tramway Jl., vol. 54, no. 1349, June 11, 1926, pp. 339–340, 3 figs. Use of three trucks for two coaches instead of each coach having two; describes such articulated cars in use in Germany and their electrical equipment.

Italy. 25 Years of Electric Operation in Italy, G. Bianchi. Elec. Ry. Jl., vol. 67, no. 24, June 12, 1926, pp. 1002–1007, 11 figs. Three-phase system has given good account of itself and has greatly increased capacity of Italian state railways; system is being extended and 300-volt d.c. lines are being built also.

ELECTRIC WELDING, ARC

Automotive Production. Automatic Arc Welding Finds Many Uses in Automotive Production, W. M. B. Brady and W. L. Warner. Automotive Industries, vol. 54, no. 25, June 24, 1926, pp. 1092–1094, 8 figs. Employed for joining universal joint to drive shaft, for frame joints, for rear-axle housings and for manufacture of various types of metal bodies; faster than hand method.

Building Construction. Arc Welding for Building Construction. Iron Age, vol. 118, no. 6, Aug. 5, 1926, p. 344. Westinghouse Electric & Mfg. Co. stages demonstration at Pittsburgh; will weld frame of 5-story building.

Crane Runways. Welded Crane Runway, A. F. Davis. Am. Welding Soc.—Jl., vol. 5, no. 5, May 1926, pp. 35–41, 7 figs. Work of Morgan Eng. Co., Alliance, O., in building great industrial crane runway without use of rivets; involved in welded job are 65 tons of steel for supporting frames and crane girders, which provide runway of 245 ft.

Direct-Current. Welding Plants for D.C. Arc Welding (Schweissanlagen für Gleichstrom-Lichtbogenschweissung), K. Meller. Elektro-Jl., vol. 6, no. 6, Mar. 25, 1926, pp. 83–91, 17 figs. Discusses developments in electric welding, static characteristics of machines, shunt-wound machines with separate or self-excitation high-frequency current superposed on welding current; transformers, etc.

Gaseous Atmosphere. Electric Arc Welding in Gaseous Atmosphere. West. Machy. World, vol. 17, no. 6, June 1926, pp. 245–246, 5 figs. By surrounding ordinary welding electrodes with atmosphere of hydrogen or certain other gases, it has been found possible to produce ductile welds; gas acts as flux and shield against oxygen and nitrogen of air, preventing formation of oxides and nitrides of iron in molten metal.

General Electric Co. Works. Arc Welding in the General Electric Plant, M. A. Atuesta. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 854–856, 5 figs. Different lines of apparatus on which arc welding is used; arc welding on motor parts and welding generator stator frames; reclaiming cast iron and steel castings; welding miscellaneous shop equipment.

Jigs and Fixtures. Arc Welded Jigs and Fixtures, E. Vom Steeg, Jr. Am. Welding Soc.—Jl., vol. 5, no. 5, May 1926, pp. 60–62, 4 figs. Making of jigs for use of commercial structural shapes, which are arc-welded together.

Machine Construction. Electric Arc Welding in Machine Construction (Lichtbogenschweissung im Maschinenbau), K. Meller. Elektrischer Betrieb, vol. 24, no. 3, Feb. 10, 1926, pp. 17–19, 20 figs. Author presents number of photographs of frames, bedplates, pipe and drum structures, and engine parts, in construction or repair of which electric arc welding is employed; in each case line of welds is shown clearly, and useful data are given concerning time occupied by various jobs; other applications of process are enumerated.

Research. Research, the Beacon of Progress in Arc Welding, H. M. Hobart and W. Sparagen. Am. Welding Soc.—Jl., vol. 5, no. 5, May 1926, pp. 13–17 and (discussion), 27–35. Problems in arc welding which require study.

Stainless Steel. Electric Welding the New Stainless Steels, C. J. Holslag. Am. Welding Soc.—Jl., vol. 5, no. 5, May 1926, pp. 57–60. Procedure is same as for iron and steel welding except that iron or steel electrode is useless and in general wire drawn for welding of material to be welded gives best results.

EMPLOYEES

Women. Effects of Applied Research Upon the Employment Opportunities of American Women. U. S. Dept. Labor—Bul. of Women's Bur., no. 50, 1926, 49 pp. Illustrative descriptions of conspicuous changes which have occurred in employment opportunities of women in certain industries as result of application of research; what these facts suggest.

ENGINEERS

Coöperative Training. Coöperative Engineering Training, F. C. French. Indus. Mgmt. (N. Y.), vol. 71, no. 6, June 1926, pp. 382–385. General scheme of coöperative training; system that tends to be selective.

F

FANS

Casings and Inlets. Experiments on Fan Casings and Fan Inlets, H. Briggs and J. N. Williamson. Mech. World, vol. 80, no. 2062, July 9, 1926, pp. 30–32, 7 figs. Function of fan casing and Rateau's diffuser; methods of measurement; velocity of air inside casing; measurement of centripetal and circumferential reentry; effects of modifying profile of casing; experiments with narrower spiral casing; experiments with five types of inlet duct; conclusions.

FILES

Manufacture. The Manufacture of Files. Mech. World, vol. 79, no. 2060, June 25, 1926, pp. 483–484, 7 figs. Describes manufacture of "stag" brand files at works of E. Allen & Co. Ltd., Sheffield, Eng.; steel used is cast steel of about 1.20 to 1.30 per cent carbon content.

Manufacture. Making Files by Modern Methods, C. O. Herb. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 951–953, 6 figs. Process in which tapered sectional miling cutter is used to mill file teeth with under-cut or lake.

FLIGHT

Soaring. The Soaring Flight Problem, H. Hartshell. Aviation, vol. 20, no. 26, June 28, 1926, pp. 975–977, 4 figs. Principles involved in soaring; soaring cycle; analysis in vertical and horizontal planes; sailing action.

FLOW OF GASES

High Speeds. On the Flow of Gases at High Speeds, T. E. Stanton. Roy. Soc.—Proc., vol. 3, no. A758, June 2, 1926, pp. 306–339, 15 figs. Experimental study of distribution of pressure and velocity in jets flowing through orifices of different forms, with special reference to existence of minimum section of jet and its variation in position and magnitude; relation between pressure in receiver and pressure in jet for increasing and constant rates of discharge, and possibility of characteristics of high-speed jets being affected by dimensions of orifice and viscosity of air; study of motion of jets at speeds above velocity of sound, and nature of wave motion set up on their emergence into quiescent air.

FLUE-GAS ANALYSIS

CO_2 Records. CO_2 Records and Their Interpretation, C. F. Wade. Elec. Rev., vol. 98, no. 2533, June 11, 1926, pp. 863–865, 1 fig. Points out that in case of large boiler units, two combustion records should be kept, first being taken at middle chamber and preferably including both CO_2 and CO charts for observation and control of air and fuel regulation, while plain CO_2 recorder installed at point of gas will, by comparison with first record, furnish evidence as to physical state of boiler settings with regard to air leakage and overall conditions.

FLYING BOATS

Supermarine Swan. The Supermarine "Swan" Commercial Flying Boat. Flight, vol. 18, no. 24, June 17, 1926, p. 344, 2 figs. Results of further flight trials at Southampton.

FOREMEN

Training. Do We Believe in Foremen Training? W. Hartman. Am. Mach., vol. 65, no. 1, July 1, 1926, pp. 1–4. Describes organized system on foremen training which has been adopted by National Cash Register Co.

Foremen Training Through Foremen's Clubs, T. B. Fordham. Am. Mach., vol. 65, no. 5, July 29, 1926, pp. 205–207, 2 figs. Need for foremen's clubs in industry and in community; advantages to foremen and to management; what has been accomplished in Ohio.

FORGE SHOPS

Locomotive Forgings. Where Alco Forgings Are Made. Ry. Rev., vol. 78, no. 24, June 12, 1926, pp. 1071–1072, 5 figs. Methods and equipment of modern forge shop at Brooks Works, Dunkirk, N. Y., of Am. Locomotive Co.

FORGING

Macrostructure Test. Flow of Metal in Forging and the Use of Macrostructure Testing (La bavure d'estampage et l'utilité des essais de macrostructure), E. Dercherf. Revue Universelle des Mines, vol. 10, no. 4, May 15, 1926, pp. 162–168, 6 figs. Discusses forging or deformation and control of flow lines or fiber lines; macrographic tests are made to reveal flow lines, etc.

FOUNDRIES

Cost Accounting. Foundry Costs, J. A. Thomas. Machy. (N. Y.), vol. 32, no. 11, July 1926, p. 874. In foundry specializing on class of work that is about uniform in weight and labor cost per unit, average cost per pound is satisfactory basis for determining selling prices; however, if foundry output is diversified, average cost would be useless.

Iron-Foundry Costing. Foundry Trade Jl., vol. 33, no. 512, June 10, 1926, pp. 430, 432 and 434. Describes, with examples, system which has been prepared and worked for considerable period.

Fuel Economy. Foundry Fuel Economies, W. J. May. Mech. World, vol. 79, no. 2058, June 11, 1926, pp. 450–451. Chief form of economy is to have good furnaces; where used, air blast should be capable of adjustment; coke should be used in as nearly exact proportions as can be managed, but amount varies with kind of furnace used.

F. K. Th. Van Iterson. Ingenieur, vol. 41, no. 17, Apr. 24, 1926, pp. 321–333, 12 figs. Calculation of coefficient of heat transmission; air preheating and preheaters; air cooling, etc.

Coaxial Cylinders, between. Heat Transfer in the Annular Space Between Two Coaxial Cylinders, S. R. Parsons. Phys. Rev., vol. 27, no. 6, June 1926, pp. 788–792. Two brass tubes are mounted with common axis, one inside of other, and are warmed by driving hot air in turbulent flow through annular space between them; purpose of work was to show whether or not heat is transmitted more readily to one of two surfaces than the other.

HEATING

Central, Steam and Water. Central Heating Plant for Canadian Government Buildings, P. H. Bryce. Heating & Ventilating Mag., vol. 23, no. 7, July 1926, pp. 78–81, 5 figs. Principal features of combination steam and water heating system for federal group in Ottawa.

Industrial. Industrial Heating by Oil Circulation, A. B. McKechnie. Gas Age-Rec., vol. 58, no. 2, July 10, 1926, pp. 35–36, 40 and 42, 2 figs. Development of Merrill process for industrial heating by oil circulation; problems encountered and how they were overcome.

HEATING AND VENTILATION

Factories. The Ventilation and Heating of Factories. Engineering, vol. 122, no. 3168, July 23, 1926, pp. 102–103. Review of data published by (Brit.) Industrial Fatigue Research Board, giving result of investigation extending over two years during which systematic observations were made in winter and in summer in 12 factories.

Madison Square Garden, New York. Mechanical Equipment of New Madison Square Garden, F. W. Thompson. Heating & Ventilating Mag., vol. 23, no. 7, July 1926, pp. 67–77, 12 figs. Details and method of operation of heating, ventilating, and cooling system.

HEATING, HOT-WATER

Cast-Iron Radiators. Dissipation of Heat by Cast-Iron Water Radiators, F. E. Giesecke. Heat. & Vent. Mag., vol. 23, no. 6, June 1926, pp. 67–72, 9 figs. Recent tests which show fallacy in use of comparatively small pipes and superiority of two-pipe over one-pipe water-treating systems.

HEATING, STEAM

Low-Pressure. The Argument for Low-Pressure Heating Plants, A. L. Sanford. Heat. & Vent. Mag., vol. 23, no. 7, July 1926, pp. 76–77. Comparison of high- versus low-pressure systems and description of typical heating installations in Minneapolis public schools.

Requirements. Analyzing Steam Heating Requirements, W. L. Berry. Elec. World, vol. 88, no. 1, July 3, 1926, pp. 5–7, 2 figs. Methods used by Union Electric Light & Power Co., St. Louis, to determine actual steam requirements and amount of exhaust steam wasted.

HYDRAULIC TURBINES

Francis. Determination of Main Dimensions of Francis Turbines (Die Bestimmung der Hauptabmessungen der Francisturbine), J. Hýbl. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 26, June 26, 1926, pp. 879–884, 8 figs. Based on examples of Francis turbines of different makes, values are derived which give suitable dimensions and designs for given conditions.

Propeller-Type. Propeller Turbine Aids Low Head Developments, J. S. Carpenter. Power Plant Eng., vol. 30, no. 12, June 15, 1926, pp. 703–704, 3 figs. High specific speeds decrease first cost and increase output of turbine at reduced heads.

Size Limitations. Developments in Hydro Electric Practice, C. R. Martin. Purdue Eng. Rev., vol. 21, no. 4, May 1926, pp. 7–10, 4 figs. Size of hydroelectric machinery is becoming limited by mechanical restrictions.

Specific Speed. The Character of the Specific Speed, B. Eck. Indian Eng., vol. 79, no. 20, May 15, 1926, pp. 278–280, 1 fig. Objections which can be raised against definition now in use; states relationship between turbines, centrifugal pumps and related engines, and shows how non-dimensional term related to specific speed can be found from physical considerations.

HYDROELECTRIC DEVELOPMENTS

Georgia. The Bartlett's Ferry Hydroelectric Development, H. A. Hageman and T. B. Parker. Stone & Webster Jl., vol. 38, no. 6, June 1926, pp. 772–805, 9 figs. Provides for ultimate generating capacity of 60,000 kw.; initial installation consists of two 15,000-kw. units; construction and equipment.

HYDROELECTRIC PLANTS

Automatic. Automatic Hydro-Electric Plant Designed for Peak Load Service, G. C. George. Power, vol. 64, no. 4, July 27, 1926, pp. 116–119, 6 figs. Installation in Soft Maple plant of Northern New York Utilities, Inc., designed for three 10,500-hp. turbines, two of which have been installed to operate under 121.5-ft. head; units are started and stopped automatically by closing or opening oil switch at another station 4½ miles distant. See also Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 858–860, 5 figs.

Brazil. Hydroelectric Plant at Cubatao of the Sao Paulo Tramway, Light & Power Co. (As obras da Grande installacao hydro-electrica de Cubatao da Companhia The Sao Paulo Tramway Light and Power.) Revista Brasileira de Engenharia, vol. 11, no. 4, Apr. 1926, pp. 139–155, 9 figs. Reservoir and dam in course of construction at Cubatao on river of same name; tunneling, pressure conduits, turbines and generator, to supply power to Sao Paulo at 80,000 volt.

Earthquakes, Effect on. Effect of Earthquake Shocks on Hydroelectric Plants (L'influence des tremblements de terre sur les installations hydro-électriques), M. Legouez. Société Française des Electriciens—Bul., vol. 6, no. 54, Feb. 1926, pp. 121–126, 2 figs. Inertia of upper stories of buildings and of masses of water must be taken into account; damage is usually due to landslides, insufficiently solid foundations, or lack of elasticity in buildings; work should not be built where there is change in geological structure of earth; details of damage to electrical apparatus in Japanese earthquake in 1923; Italian rules for constructing foundations.

Germany. Inn Plant Helps Industrial Germany. Power Plant Eng., vol. 30, no. 13, July 1, 1926, pp. 760–761, 3 figs. Details of Innwerk, hydro-electric plant on River Inn near Toeging, Upper Bavaria; total capacity of 98,600 hp. over 465,000,000 kw-hr. will be produced annually; from dam concrete tunnel 12.5 miles long carries water to power house; turbines are of cased type employing single, flat-curved-bladed, reaction runners with inlet parallel to shaft.

Operating Routine and Tests. Hydro Plant Operating Routine and Tests. Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 860–862, 2 figs. Organization of force, keeping of records and layout of operating schedules must be considered.

Russia. The Kondopoga Hydroelectric Station, P. G. Skoolsky. Electritchestvo, no. 5, May 1926, pp. 253–256, 9 figs. Gives history and describes erection of station on Soona River in North Russia. (In Russian.)

Scotland. The Clyde Valley Electric Power Co. Elec. Rev., vol. 98, no. 2535, June 25, 1926, pp. 944–947, 10 figs. Developments of Clyde Valley Power Co. in vicinity of Glasgow, Scotland. See also Elecn., vol. 96, no. 2506, June 11, 1926, pp. 598–599 and 603, 6 figs.

I

ICE MANUFACTURE

Sub-Cooled Raw Water. Reversing the Ice Making Process, C. H. Herter. Power Plant Eng., vol. 30, no. 12, June 15, 1926, pp. 710–711. Discussion of feasibility of making ice from sub-cooled raw water.

ICE PLANTS

Distilled-Water. A Modern Distilled Water Compression Plant. Refrigeration, vol. 38, no. 6, June 1926, pp. 39–42, 3 figs. Steam-driven distilled-water plant erected by Morgan Ice & Coal Co., Knoxville, Tenn.

Pipe Insulation. Ground Corn Cobs for Pipe Line Insulation, G. G. McVicker. Nat. Engr., vol. 30, no. 7, July 1926, pp. 309–310, 3 figs. How ground corn cobs were used for insulating brine-circulating pipes in ice plant; making form to receive insulation; applying insulation and cost of materials.

INDUSTRIAL MANAGEMENT

Machine Shops. Economic Shop Management (Ny ökonomisk verkstedsdrift), C. Munthe. Teknisk Ukeblad, vol. 73, nos. 11, 12 and 14, Mar. 12, 19 and Apr. 9, 1926, pp. 98–99, 108–113 and 127–131, 23 figs. Discusses modern machine-shop practice, coöperation between office and shops, specializing and limitation of types of products; keeping machines employed; systematic management of labor; shop equipment, machines and tools, cost accounting and wages.

Material-Order System. A Simple Material-Order System for the Small Plant, C. F. Shafer. Factory, vol. 37, no. 1, July 1926, pp. 53 and 88, 1 fig. Steps in the system of ordering materials for the small plant.

Planning. Planning and Routing Shop Work, J. S. Gray. Machy. (Lond.), vol. 28, no. 715, June 24, 1926, pp. 342–344. Work of planning department; effect of planning on cost and delivery dates; coöperation between shop and planning department and advantages to be gained by proper planning.

Production Control. A New Electrical Production Control System, L. Ederer. Soc. of Indus. Engrs.—Bul., vol. 8, no. 6, June 1926, pp. 3–5. Electrical control equipment consisting of series of substations placed at convenient points throughout plant and connected electrically to central station in such a manner that information is sent directly from substations and recorded at central station.

The Factory Manager Speaks on Production Control and Safety, W. E. Irish. Am. Mach., vol. 64, nos. 19 and 20, May 13 and 20, 1926, pp. 735–736 and 785–787. May 13: Methods of production control based on premise that system shall fit itself to work in hand and grow up from very simple beginning. May 20: Correcting speeds and feeds; changing price on piece work; group payment for progressive piece-work assembly.

The Problem of Maintaining Production, J. S. Gray. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 968–969. Providing materials for production; methods of moving materials; scheduling and planning work; coöperation between production department and foremen; watching out for excessive spoilage, scrap and waste; effect of wages on production.

Thousands of Orders: No Late Deliveries, R. J. Rahn. Factory, vol. 36, no. 6, June 1926, pp. 1046–1047 and 1082, 3 figs. Simple produticon-control board for plant that manufactures to order.

Production Planning. Factors in Industrial Organization, R. Wendorby. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, pp. 251–252. Author divides controlling factors into five groups, namely, intelligent buying, centralized storage and expeditious distribution, efficient production, prompt and reliable inspection, and rapid transit.

Purchasing. Purchasing as a Function of Management, C. A. Merrill. Ry. Age, vol. 81, no. 1, July 3, 1926, pp. 21–22, 2 figs. Present supply problems emphasize importance of specialized training in this work.

Selling-Price Calculation. Are Profit Ideals High Enough? K. H. Condit. Am. Mach., vol. 65, no. 5, July 29, 1926, pp. 189–191. Collection of statements by leading industrial executives giving their ideas on how selling prices should be established.

Shop Records. Shop Records that Increase Efficiency, J. S. Gray. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 857–859. Cost records must be made to determine not only whether profit is being made on product, but also whether cost of manufacturing is increasing or decreasing; effect of records on organization; when department foremen should keep records of their own; how to file records.

Stock Keeping. Profit and Loss in Stock Keeping, R. B. Williams. Can. Machy., vol. 36, no. 2, July 8, 1926, pp. 10–12, 6 figs. Basic requirements; stock-keeping system; identifying each shelf; storing plate patterns; adequate protection.

Toolrooms. Tool Room System. Automobile Engr., vol. 16, no. 216, June 1926, pp. 214–216, 6 figs. Practical method of progressing and costing in tool making.

INDUSTRIAL ORGANIZATION

Improvements. Industry's Silent Revolution—and What It Portends, M. W. Alexander. Iron & Trade Rev., vol. 78, no. 25, June 24, 1926, pp. 1622–1625, 4 figs. Effects upon industry of rapid increase of population; improvements in organization and in technical processes; increase in influence of corporately owned establishments, etc.

INDUSTRIAL PLANTS

Relocation. The Effects of Industrial Plant Relocation, J. A. Fiquet. Indus. Mgmt. (N. Y.), vol. 71, no. 6, June 1926, pp. 376–380, 7 figs. Physical and psychological results.

INDUSTRIAL RELATIONS

Future Prospects. Future Relationship Between Employer and Employed. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, p. 247. Symposium of opinions received from leaders of industry on future relationship between employer and employed and its bearing on industrial reconstruction.

Personnel, Study of. Influence of Personnel on Industry, R. A. C. Henry. Eng. Jl., vol. 9, no. 7, July 1926, pp. 333–336. Necessity of studying personnel in industry and possibility of creating better relationship between employer and employee.

INDUSTRIAL TRUCKS

Electric. The Expanding Uses of the Electric Industrial Truck, H. J. Payne. Factory, vol. 36, no. 6, June 1926, pp. 1035, 1130, 1134 and 1138. Comprehensive report shows opportunities for use of electric trucks in many industries not yet familiar with them and for their use in new ways by present owners.

INSULATORS, HEAT

Zonolite. Zonolite, a New Light-Weight Insulator, J. Keeth. Rock Products, vol. 29, no. 13, June 26, 1926, pp. 63–64, 1 fig. New insulating material called zonolite which weighs only 7 lb. per cu. ft. and has good fire-resistant qualities.

INSURANCE

Group Life. Making Life Insurance Easy for Employees, J. J. Birch. Indus. Mgmt. (N. Y.), vol. 71, no. 6, June 1926, pp. 349–350. General Electric Company has recently inaugurated group-life and salary-allotment insurance plan for optional use for its employees; types of salary-allotment policies; total permanent-disability benefits.

INTERNAL-COMBUSTION ENGINES

Portable. Light Engines for Special Purposes (Der Leichtmotor für Sonderzwecke), K. R. H. Praetorius. Automobil-Rundschau, vol. 28, nos. 5 and 6, May 1 and June 1, 1926, pp. 107–111 and 137–138, 14 figs. Application of small-size engines for driving small radio transmitter stations, machine tools, etc.; types built by Siemens & Halske, Bergmann, etc., from 1½ hp. upwards.

Supercompression. Supercompression in Explosion Engines (La surcompression dans les moteurs a explosion), M. Dumanois. Technique Moderne, vol. 18, no. 9, May 1, 1926, pp. 257–261, 2 figs. Discusses effect of increased compression on combustion, and efficiency; spontaneous ignition, detonation; their causes and effects.

Tests. Industrial Tests of Internal Combustion Engine, W. A. Tookey. Diesel Engine Users Assn., no. S70, for mtg. Apr. 9, 1926, 40 pp. 11 figs. Tests made by manufacturers under supervision of engineer acting on behalf of purchaser; preliminary arrangements, duration and scope of trials; power output; fuel consumption; governing and speed variation; control; thermal and mechanical efficiencies.

[See also AIRPLANE ENGINES; AUTOMOBILE ENGINES; DIESEL ENGINES; GAS ENGINES; OIL ENGINES.]

IRON

Rustproofing with Lead. Rustproofing with Lead (Verbleiung durch Anstrich), A. V. Blom. Zeit. für angewandte Chemie, vol. 39, no. 18, May 6, 1926, pp. 555–556, 2 figs. By a new method it is possible to spread lead coating on iron like paint; pyrosol of lead in lead oxide is made by heating lead in electric furnace under closely regulated conditions; product, very finely powdered, is then dispersed in linseed oil; spread

on clean iron surface, this pigment yields metallic lead in highly activated state and film of lead forms on iron; coating is extraordinarily adherent.

IRON ALLOYS

Constitution. The Constitution of Steel and Cast Iron, F. T. Sisco. Am. Soc. for Steel Treating—Trans., vol. 9, no. 6, June 1926, pp. 938–953, 11 figs. Discusses fundamental concepts necessary to complete understanding of constitution and structural changes of alloys of iron and carbon; included in fundamentals are simple crystallization, saturation and equilibrium, crystallization of metals, orientation and cleavage, crystal structure, amorphous solids, allotropy and allotropy of iron.

IRON AND STEEL

Macroscopic Examination of. Macroscopic Examination of Iron and Steel, F. P. Gilligan and J. J. Curran. Am. Soc. for Steel Treating—Trans., vol. 10, no. 1, July 1926, pp. 9–25 and (discussion) 25–30, 21 figs. Describes deep-etch method of testing or inspecting bars, billets, or finished parts; discusses procedure in using this method of test, merits of various acid mixtures used in detecting different types of defects occurring in steels which may be revealed by this type of test; comparison between this and microscopic test is made; photographs are shown of specimens tested by this method and nature of defects as revealed by this method are described.

IRON CASTINGS

Gates and Runners. Arrangement of Gates and Runners in Gray-Iron Founding. (Anordnung der Trichter und Eingüsse bei Grauguss) Zeit. für die Gesamte Giessereipraxis, vol. 47, nos. 18 and 19, May 2 and 9, 1926, pp. 201–203 and 213–215, 33 figs. Gives illustrations showing arrangements for various classes of work.

Large. Large Castings Used. Foundry, vol. 54, no. 12, June 15, 1926, pp. 456–460, 6 figs. Methods and equipment of Dominion Engineering Works, Montreal, Can., in production of large castings for papermaking machinery, hydraulic turbines, electric steam generators, etc.

IRON FOUNDING

Research. The Value of Research in the Production of Castings, J. G. Pearce. Mech. World, vol. 79, nos. 2041 and 2049, Feb. 12 and Apr. 9, 1926, pp. 129–130 and 289–290. Feb. 12: Shows value of research in iron founding; how every-day difficulties may be explained and corrected by metallurgical knowledge and experience; and how current research work is resulting in developments which call for serious attention of industry. Apr. 9: Deals with Emmel iron, hot-mold, Schuz and Nomag iron.

IRON, PIG

Cylinder. Some Notes on the Production of Cylinder Pig-Iron to Fracture and Analysis, E. J. Yates. Foundry Trade Jl., vol. 33, no. 513, June 17, 1926, pp. 457–458. Outlines difficulties and experiences on production of cylinder pig iron from metallurgical and technical points of view.

Remelting. Re-Melting Pig-Iron in the Open-Hearth Process, E. Herzog. Foundry Trade Jl., vol. 33, no. 514, June 24, 1926, pp. 500–501. Describes cupolas used at Rothe Erde Works, Germany; difficulties encountered and how they were overcome.

J

JAPANNING

Ovens. Japanning Stove Parts in a Continuous Conveyor Oven, W. E. Crum. Fuels & Furnaces, vol. 4, no. 7, July 1926, pp. 829–831, 3 figs. Oven is of double-tunnel type, each tunnel ventilated separately and provided with conveyor driven by individual motors giving independent control of conveyor speeds.

L

LATHES

Aprons. Lathe Aprons. Brit. Machine Tool Eng., vol. 3, no. 39, May-June 1926, pp. 413–417, 8 figs. Various designs of Lang lathe apron produced to bring feed-engaging mechanism of that unit to its present state of efficiency.

Crankshaft. 34-Inch Crankshaft Lathe Is Added to Wickes Line. Automotive Industries, vol. 54, no. 25, June 24, 1926, pp. 1101–1102, 3 figs. New model is of semi-automatic type with silent chain drive from 15–20-hp. motor; eight changes of feed obtainable; lever used for starting spindles also serves as brake to stop them.

Turret. An Improved Hexagon Turret Lathe. Engineer, vol. 141, no. 3675, June 18, 1926, pp. 642–644, 10 figs. Improved design of lathe which is product of A. Herbert of Coventry, Eng.

LIGHTING

Industrial. Industrial Lighting (L'éclairage dans l'industrie), M. J. Wetzel. Electricité & Mécanique, no. 11, Mar.-Apr. 1926, pp. 32–46, 20 figs. Discusses relation between lighting and eyesight, quicker accommodation of eye in good lighting, increased production, reduction of accidents.

Plant Illumination, W. Harrison. Engrs. & Eng., vol. 43, no. 6, June 15, 1926, pp. 149–151, 3 figs. Author refers to practical tests which have been made in number of industries and which show that supplying more light increases production and decreases unit cost.

LOCOMOTIVE BOILERS

Firing and Care. Firing and Care of Locomotive Boilers, J. A. Good. Eng. & Boiler House Rev., vol. 39, no. 12, June 1926, pp. 585–586. Practical suggestions for maintenance of efficiency. (Abstract.) Paper read before Assn. of Hawaiian Sugar Technologists.

McClellon. New Type Boiler on a Three Cylinder Engine. Ry. Rev., vol. 78, no. 24, June 12, 1926, pp. 1061–1066, 10 figs. Successful trials of both principles by New Haven lead to incorporation in new power.

LOCOMOTIVES

Central Railroad of New Jersey. Locomotive Development on the Central Railroad of New Jersey, P. T. Warner. Baldwin Locomotives, vol. 5, no. 1, July 1926, pp. 3–25, 56 figs. Traces development from 1896 to present time.

Design. Report on Locomotive Design and Construction. Ry. Age, vol. 80, no. 24, June 15, 1926, pp. 1819–1829, 7 figs. Standardization of taps and dies; bolt and screw threads; designing locomotives to reduce track stresses; standardization of water columns; definition of engine failure. Report of committee before Am. Ry. Assn.

Diesel-Electric. Diesel Locomotive with Gear Transmission, B. A. Wittkuhus. Ry. Mech. Engr., vol. 100, no. 7, July 1926, pp. 425–428, 8 figs. Describes two Diesel-electric locomotives designed for freight service, one is 2-10-2 type equipped with 1200 b.hp. M.A.N. Diesel engine, and other is 4-10-2 type with gear-transmission system equipped with same engine.

Electric. See ELECTRIC LOCOMOTIVES.

Exhibits, Atlantic City. Locomotive Exhibit Shows Development. Ry. Age, vol. 80, no. 23, June 14, 1926, pp. 1772–1776, 4 figs. High-pressure and three-cylinder locomotives of advanced design; internal-combustion and electric locomotives included.

Internal-Combustion. Internal Combustion Locomotives and Vehicles, S. M. Vauclain. Baldwin Locomotives, vol. 5, no. 1, July 1926, pp. 43–49, 8 figs. Types of railway motor cars; D locomotives, with mechanical, hydraulic and electric transmission.

Mountain Type. 4-8-2 Compound Locomotives of the P.L.M. Railway. (Locomotive Compound A Quatre Essieux Couplés de la Compagnie des Chemins de fer P.L.M.), Génie Civil, vol. 88, no. 24, June 12, 1926, pp. 517–519, 9 figs. Particulars of new 117-ton compound superheater locomotive for express service between Paris and Dijon.

Multiple-Pressure. Multiple Pressure Locomotive Developed in Germany. Boiler Maker, vol. 26, no. 6, June 1926, pp. 152–153, 3 figs. Also Ry. Age, vol. 80, no. 38, June 26, 1926, pp. 1969–1970, 3 figs. Locomotive, known as Henschel multiple-pressure compound locomotive, is of 4-6-0 type and has 3 cylinders; combined water-tube and fire-tube boiler generates steam at 800- and 200-lb. pressure.

Narrow-Gage. Indian Locomotives for Narrow Gage (Indische Schmalspur-Lokomotiven), Metzeltin, Organ für die Fortschritte des Eisenbahnwesens, vol. 81, no. 9, May 15, 1926, pp. 165–168, 7 figs. Details of various types of locomotives, built by Kerr, Stuart & Co., Yorkshire Engine Co., and Hanomag.

Pistons, Standardization of. Standardization and Maintenance of Steam Pistons and Piston Rings in German Locomotives (Die Normung und Unterhaltung der Dampfkolben und Kolbenringe bei den vorhandenen Reichsbahn-Lokomotiven), Glasers Annalen, vol. 98, no. 11, June 1, 1926, pp. 174–177, 5 figs. Shows that by adopting degrees of wear and fixed dimensions for shrunk rings maintenance of pistons in repair works may be facilitated and reduced in cost.

Pressure Gages. Engine Operation Standardized by Pressure Gages. Ry. Age, vol. 80, no. 37, June 19, 1926, pp. 1903–1905, 2 figs. Union Pacific experience in recent months with steam chest and back-pressure gages on locomotives shows important fuel saving.

Rail-Stress-Reducing. Designing Locomotives to Reduce Rail Stresses, H. H. Lanning. New England Railroad Club—Official Proc., Apr. 13, 1926, pp. 78–93 and (discussion) 93–101. How Atchison, Topeka & Santa Fe Railway Co. built heavier and more powerful locomotives without overtaxing strength of existing track and bridges; describes four standard types of steam locomotives adopted.

Single-Expansion. Single Expansion Articulated Locomotives in Mountain Service on the Great Northern Railway, H. Blanchard. Baldwin Locomotives, vol. 5, no. 1, July 1926, pp. 52–56, 4 figs. Details of 2-10-2 and 2-8-2 types.

Superheater. 1 C Superheater Tender Locomotives for Meter-Gage Railways (1 C-Heissdampf-Tenderlokomotive für meterspurigs Kleinbahnen). Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 25, June 19, 1926, p. 41, 2 figs. Type developed by German Gen. Elec. Co. (A.E.G.).

Tank. 2-6-4 Tank Locomotives for Indian Railways. Ry. Gaz., vol. 44, no. 26, June 25, 1926, pp. 812–813, 4 figs. Describes two heavy tank engines of 2-6-4 type to be used on Indian railways.

Tenders. Tenders of Large Capacity for the Great Northern. Ry. Mech. Eng., vol. 100, no. 7, July 1926, pp. 429–430, 5 figs. Seventeen locomotive tenders largest that have ever been built for regular railroad service; have coal capacity of 24 tons and water capacity of 21,500 gal.; light weight is 138,100 lb. and loaded 384,550 lb. or 182 tons; have rigid anchorage between body and underframe.

Three-Cylinder. Three-Cylinder Locomotive Performance Record. Ry. & Locomotive Eng., vol. 39, no. 6, June 1926, pp. 153–154, 1 fig. Locomotive No. 5000 of Lehigh Valley Railroad shows low maintenance cost per locomotive mile; it is 4-8-2 type and weighs 369,000 lb.

Union Pacific. 4-12-2 Locomotive for Union Pacific. Ry. Mech. Engr., vol. 100, no. 7, July 1926, pp. 419–424, 12 figs. Built for fast freight service in mountain districts; has 67-in. drivers and develops rated tractive force of 96,650 lb.

Weight Distribution. The Distribttion for Locomotive Weights. Ry. & Locomotive Eng., vol. 39, no. 6, June 1926, pp. 154–158, 13 figs. Survey of changes in distribution of weights on wheels after locomotives have been in service.

LUBRICATING OILS

Used, Treatment of. The Treatment of Used Oil (Die Aufbereitung gebrauchter Oele), v. d. Heyden and Typke. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 12, Mar. 20, 1926, pp. 401–402. Considerable savings may be effected by treating used lubricating and other oils, so as to eliminate impurities and correct changes in composition caused by service conditions; treatment with Fuller's earth removes mechanical impurities and adsorbs acids and dissolved asphaltic matter; oil may either be passed through filter bed of granular Fuller's earth, or agitated with finely divided earth in stirring machine; latter method is more effective.

LUBRICATION

Diesel Engines. Lubricating Oils for Heavy-Oil Engines. Marine Engr. & Motorship Bldr., vol. 49, no. 587, July 1926, p. 252. Particulars of report issued by committee of Diesel Engine Users Association on specification of such oils.

Materials-Handling Equipment. Lubrication of Materials Handling Equipment. Power Plant Eng., vol. 30, no. 12, June 15, 1926, pp. 700–702, 5 figs. Proper selection of lubricant to fit working conditions and proper application to gears, bearings and wire rope increase efficiency.

Oil Filtration. Lengthening the Life of Oil. Autocar, vol. 57, no. 1660, July 2, 1926, p. 33, 2 figs. Description of Hele-Shaw streamline lubricant filter, and method of applying it to automobile engines.

M

MACHINE TOOLS

Developments in. Shop Equipment Review. Am. Machinist, vol. 65, no. 3, July 15, 1926, pp. 77–149, 369 figs. Semi-annual résumé of some 350 American and European machines, tools and accessories described in shop-equipment news section of Am. Machinist during first six months of 1926.

High-Production. High-Production Machine, E. J. Bullard, Jr. Am. Mach., vol. 65, no. 6, Aug. 5, 1926, pp. 229–231. Points out that modern high-production machines are based on principles of machine tool, but their successful design to meet competitive conditions must also incorporate basic principles of good management, which include reduced space, less skilled labor, smaller work-in-process inventories, reduced handling time, and maximum production.

Increased Production from. The Production Problem, G. W. Rawlings. Mach. Market, no. 1340, July 9, 1926, pp. 19–20, 4 figs. Features of machine tools responsible for increased production; gear planers; boring machines.

MANOMETERS

Thermoelectric. Thermoelectric Manometers for Small Pressures (Ueber ein thermoelektrisches Manometer für kleine Drücke), E. Rumpf. Zeit. fur Technische Physik, vol. 7, no. 5, 1926, pp. 224–226, 2 figs. Describes manometer consisting of heating filament and thermocouple which enables quick continuous measurement of pressures.

MARINE BOILERS

History and Development of. History and Development of the Marine Boiler, W. M. McFarland. Marine Eng. & Shipping Age, vol. 31, no. 7, July 1926, pp. 409–410. Outline of progress made in construction of Scotch-type and water-tube boilers for marine installations.

Water-Tube "Yarrow" Type. Some Recent Modifications to Watertube Boilers of the Three-Drum Type Fitted in H. M. Navy, L. M. Hobbs. Marine Eng. & Shipping Age, vol. 31, no. 7, July 1926, pp. 403–407, 8 figs. Description of alterations made as result of experiences during war; steam drums; water drums; tubes and tube distortion.

MATERIALS HANDLING

Machinery, Evolution of. Material Handling Machinery and Its Evolution, T. L. Messiter. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, pp. 240–242, 1 fig. Diagrammatic representation of evolution of handling machinery.

Port of Hamburg. Materials-Handling Installations in Hamburg Harbor (Die mechanische Umschlaganlagen im Hamburger Hafen), Wundram. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 24, June 12, 1926, pp. 826–831, 10 figs. History of development with regard to design, efficiency, safety and economy; quay revolving cranes; winches; heavy-load cranes; electric trucks; weighing equipment; loading bridges; car dumpers; progress in mechanization.

Safety in. Safe Handling of Materials, E. R. Farra. Safety Eng., vol. 51, no. 6, June 1926, pp. 343–347. Classification of accidents; correct way to lift materials; responsibility of foremen; inspection of cables; desirability of substituting mechanical methods of handling materials.

Stamping Shop. Handling Materials in a Stamping Shop, P. J. Edmonds. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 847–851, 13 figs. Systematic methods used in large shop to handle raw material, finished product, and scrap.

Standardization. Simplification and Extension of Conveying Plants by Standardizing, V. Hanffstengel. Eng. Progress, vol. 7, no. 6, June 1926, pp. 168–171, 5 figs. Advantages of standardization in conveying plants; how to determine and link up standardized measures.

MERCURY-VAPOR PROCESS

Superiority of. The Mercury-Steam Cycle, P. M. Shen. Power, vol. 64, no. 1, July 6, 1926, pp. 8–11, 2 figs. Binary-vapor conception considered as possible means of increasing efficiency; why mercury-steam cycle is superior to other binary-vapor processes.

METAL WORKING

Automobile Fenders. Making Automobile Fenders with Hand Tools, E. Heller. Machy. (N. Y.) vol. 32, no. 11, July 1926, pp. 871–874, 9 figs. Methods employed in making set of fenders for first car of new model.

Window Sash. Making Metal Window Sash, A. Bartusch. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 940–942, 7 figs. Dies and methods employed in making window sash and casements from T-section steel bars.

METALLOGRAPHY

Study of. The Engineer and the Study of Metals, W. R. Needham. World Power, vol. 5, nos. 30 and 31, June and July 1926, pp. 303–309 and 29–32. June: Discusses facilities available for study of metal, such as X-ray examination of metals; recording internal forging or ingot flaws; magnetic analysis, micro-stress recorder, photoelastic method of determining stresses, etc. July: X-ray examination of materials; clink detector for recording internal forging or ingot flaws; quest for flaws by magnetic analysis; trepanning tool for removing central cores; drop-of-potential method for detecting flaws; micro-stress recorder; photoelastic stress analysis; commercial application of light alloys for forgings; castings, etc.

METALS

Elasticity and Melting Point. Elasticity and Melting Point (Elastizität und Schmelzpunkt), J. P. Andrews. Physikalische Zeit., vol. 27, no. 7, Apr. 1, 1926, pp. 210–211. Widder's linear relation in elasticity temperature curve of metals requires line when produced to zero elasticity to cut temperature axis at melting point; expression leads to erroneous melting-point values. See also brief abstract in Brit. Chem. Abstracts, May 1926, p. 462.

Elasticity of. Impact Strength and Dynamic Elastic Limit (Dauerschlagfestigkeit und dynamische Elastizitätsgrenze), G. Welter. Zeit. des Vereines deutscher Ingenieure, vol. 70, nos. 20 and 23, May 15 and June 5, 1926, pp. 649–655 and 772–776, 76 figs. Describes new simplified test method for determining endurance strength, and values obtained for metals and alloys are compared with those obtained from tests with pendulum impact machine; investigation was carried out on iron and steel, brass, copper, aluminum, two aluminum alloys and on glass; based on results, theories on causes of permanent fracture are critically reviewed, and mechanistic theory, based on pure elasticity of materials, is shown to hold true; these investigations are said to afford a reliable guide for utility of materials under permanent stress.

Fatigue. Stress-Strain-Cycle Relationship and Corrosion-Fatigue of Metals, D. J. McAdam, Jr. Am. Soc. for Testing Matls.—Preprint, no. 33, for mtg., June 21, 1926, 31 pp., 12 figs. Material and methods of investigation; stress-cycle graph extended to nominal stresses well above tensile strength of material and over range from less than 1000 to more than 100,000,000 cycles; interrelationship between stress-cycle, stress-deflection and deflection-cycle graphs; effect of cycle repetition on deflection and hysteresis; evidence is presented that very slight corrosion when simultaneous with fatigue causes low resistance to fatigue; damaging effect is greater the harder the steel; advantage of heat treatment may thus be practically neutralized; ratios of corrosion-fatigue limits to endurance limit are compared with other physical properties of steel.

Heat Conductivity. Thermal Conductivity of Certain Metals and Alloys. (Conductibilité thermique de certains métaux et alliages), M. Jakob. Chaleur & Industrie, vol. 7, no. 74, June 1926, pp. 301–306, 1 fig. Data on conductivity of certain bronzes, aluminum, aluminum alloys, and precious metals.

Soft, Hardness of. The Hardness of Soft Metals (Die Härte weicher Metalle), R. Baumann. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 12, Mar. 20, 1926, pp. 403–404, 1 fig. Impact hardness test is only test which gives definite values; Wüst's interpretation of impact hardness.

Stress-Strain Curves. Stress Strain Curves and Physical Properties of Metal with Particular Reference to Hardness, H. P. Hollnagel. Am. Soc. for Steel Treating—Trans., vol. 10, no. 1, July 1926, pp. 87–108, 10 figs. Physical properties of metals as associated with stress-strain curve interpreted in terms of atomic forces of restitution, slip of atomic nature and in mass; interfering material set forth as amorphous phase; hardness defined; dimensions of hardness.

Plasticity. On the Plasticity of Metals, H. Shôji. Inst. of Physical & Chem. Research—Sci. Papers, vol. 4, nos. 57–58, Apr. 1926, pp. 189–201, 26 figs. Definition; description of apparatus for experiments made; plasticity of metals; change of plasticity with straining time.

On the Plasticity of Metals at High Temperatures, J. Shôji and Y. Mashiyama. Inst. of Physical & Chem. Research—Sci. Papers, vol. 4, nos. 57–58, Apr. 1926, pp. 202–205, 6 figs. Apparatus; results of experiments.

MILLING MACHINES

Oil-Groove. Lehmann Oil-Groove Milling Machine. Machy. (N. Y.), vol. 32, no. 11, July 1926, p. 913, 1 fig. Machine designed primarily for use in railroad shops for milling oil-grooves in locomotive crown brasses, but which may also readily be adapted to other work.

Repetition Work. A New Milling Machine for Repetition. Brit. Machine Tool Eng., vol. 3, no. 39, May-June 1926, pp. 428–430, 1 fig. Parkinson plain milling machine meets need for rigid high-production machine; easily controlled and capable of giving efficient speeds and feeds for any job within its capacity.

Tooling Production. Tooling Production Milling Machines. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, pp. 971–973, 7 figs. Work-holding fixtures and cutters designed to give high output rates with standard machines.

MOLDING MACHINES

Increasing Production. Increasing Output with Molding Machines, R. Micks. Can. Foundryman, vol. 17, no. 6, June 1926, pp. 17–19, 2 figs. Points out that rapid development of molding-machine practice in past few years has made it possible to produce molds by machinery for almost any casting that can be made in 2 part flask; types of molding machines.

Jarring. Jarring Machines (Machines à mouler à sécousses), A. Michel. Fonderie Moderne, no. 20, May 1926, pp. 115–118, 9 figs. Describes series of machines produced by Leber & Brose at Coblenz.

MOLDS

Permanent, for Oil Rings. Permanent Mould for Oil Rings. Machy. (Lond.), vol. 28, no. 712, June 3, 1926, p. 241, 2 figs. Method adopted for producing small oil rings in zinc alloy, by means of hand ladling metal into steel mold.

MOTOR TRUCKS

Crude-Oil. Crude Oil Automotive Engine Installed in Four-Ton Truck, L. M. Fanning. Oil & Gas Jl., vol. 24, no. 51, May 13, 1926, pp. 92 and 94. Motor truck of 4-ton capacity operating in New York City traffic on 24 deg. Baumé fuel oil; equipped with French Peugeot crude-oil engine, which can work on any kind of oil, including heaviest; it is an explosion engine in which fuel is injected and is of constant volume combustion type. See also description in Oil-Engine Power, vol. 4, no. 7, July 1926, pp. 403–405, 4 figs.

Operation Costs. How to Find Unit Costs in Motor Truck Operation, A. J. Peel. Indus. Mgmt. (N. Y.), vol. 71, no. 6, June 1926, pp. 346–349, 3 figs. Outlines of simple methods of finding unit costs which, put into operation, shows way to substantial savings.

Rolled Steel Wheels. Rolled Steel Truck Wheels. Iron Age, vol. 118, no. 1, July 1, 1926, pp. 9–11, 7 figs. Describes manufacture of rolled steel truck wheels by Bethlehem Steel Co. and laboratory tests made by Bureau of Standards.

N

NICKEL ALLOYS

Nickel-Chromium. Heat-Resisting Nickel Chrome Alloys. Foundry Trade Jl., vol. 33, no. 512, June 10, 1926, pp. 411–412, 1 fig. Research has shown that several combinations of these two elements can be effected which will withstand most arduous conditions of industrial service.

NON-FERROUS METALS

German Developments. Recent Development of Non-Ferrous Metals, H. Groeck. Eng. Progress, vol. 7, no. 6, June 1926, pp. 158–159. Discusses German supply of non-ferrous metals, and endeavors since war to find substitutes for those no longer available; characteristics of some non-ferrous metals, including copper and brass, and aluminum and its light alloys.

O

OIL

Specific Heat. Specific Heats of Oils, A. R. Fortsch and W. G. Whitman. Indus. & Eng. Chem., vol. 18, no. 8, Aug. 1926, pp. 795–800, 7 figs. Specific heats of 15 petroleum oils have been determined over temperature intervals varying from 50 to 430 deg. fahr.

OIL ENGINES

Cooling-Water Recirculation. Re-Circulating Oil Engine Cooling Water, A. B. Newell. Marine Eng. & Shipping Age, vol. 31, no. 7, July 1926, pp. 397–398. System of oil-engine cooling designed to maintain constant temperatures under all conditions of sea-water temperature.

Lubrication. The Choice of a Lubricating Oil, A. J. Wilson. Gas & Oil Power, vol. 21, no. 250, July 1, 1926, pp. 213–214. Physical tests to be considered when making choice of lubricating oil for use in gas or oil engine; specifications for oil for use on heavy-oil engines.

Marine. Marine Oil Engine Trials. Engineering, vol. 121, nos. 3150, 3151 and 3153, May 28, June 4 and 18, 1926, pp. 639–641, 675–677 and 742–744, 26 figs. Abstract of 4th report of Committee to Instn. Mech. Engrs., dealing with results of tests on M.S. British Aviator and her engines; covers shore and sea tests.

Plenty-Still. The "Plenty-Still" Oil Engine. Shipbldr., vol. 33, no. 191, July 1926, pp. 349–352, 13 figs. Explanation of Still thermal cycle; description of Plenty Still engine with details of parts.

The "Plenty-Still" Oil Engine. Mar. Engr. & Motorship Bldr., vol. 49, no. 586, June 1926, pp. 211–215, 13 figs. Experimental single-cylinder unit at 130–150 b.hp. by Plenty & Son, Ltd., Newbury; two-stroke internal-combustion cycle, and steam distribution by cam-operated poppet valves.

Sketches and Working. Sketches and Working of Oil Engines. Motorship, vol. 11, no. 7, July 1926, pp. 532–536 and 544, 14 figs. Circle diagrams aid study of reversing systems; analysis of mechanisms found in practice.

Waste-Heat Utilization. Reclaiming Waste Heat From Modern Oil Engines, A. B. Newell. Nat. Engr., vol. 30, no. 7, July 1926, pp. 301–302. Approximately one-eighth of original heat in fuel used in oil engine is available for heating purposes after having passed through engine; factors to be considered when installing equipment of this kind.

Werkspoor. The North Eastern-Werkspoor Double-Acting Engine. Mar. Engr. & Motorship Bldr., vol. 49, no. 586, June 1926, pp. 207–210, 4 figs. First engine of 4000 b.hp. completes trials at works of North Eastern Marine Eng. Co.; to be installed in single-screw motorship *Stentor*.

OIL FUEL

Burners. Test Data on Oil Bruner. Elec. World, vol. 88, no. 1, July 3, 1926, pp. 7–8. Results of tests conducted by Phila. Elec. Co. giving data on boiler efficiency and costs of oil burner for use with house furnace; burner makes desirable load for central stations.

The Development of Atomization Methods for Oil Burners, S. D. Rickard. Heat. & Vent. Mag., vol. 23, no. 6, June 1926, pp. 76–78, 3 figs. Considerations affecting flame characteristics and economies derived from experiences of past.

Handling Equipment. Fuel Oil Handling Equipment, C. C. Hermann. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 883–885, 4 figs. In planning fuel-oil system for forge-shop use, special attention should be given to certain parts of equipment; strainer for oil-pipe line; hose for tank connection; unloading station plans; oil-pump installation.

OIL SHALES

North Carolin. Oil-Bearing Shales in North Carolina, F. C. Vibrandt. Indus. & Eng. Chem., vol. 18, no. 8, Aug. 1926, pp. 793–795. Survey of Deep River Valley oil-bearing shales together with their retort assays, shale analyses, tonnage yields, and analytical distillation data of oils.

OPEN-HEARTH FURNACES

Regenerative. Regenerative Pusher Type Furnace for Heating Blooms. Fuels & Furnaces, vol. 4, no. 7, July 1926, p. 807, 2 figs. Describes regenerative continuous furnace of pusher type built in Germany for heating blooms $6\frac{1}{4}$ in. $\times$ $6\frac{1}{4}$ in. cross-section and 86 in. long, weighing 880 lb.

OXYACETYLENE CUTTING

Ship Scrapping. The Ship Scrapping Industry. Welding Engr., vol. 11, no. 6, June 1926, pp. 41–42, 4 figs. Contractors find various ways to reduce costs of cutting up steel ships and similar structures with oxyacetylene cutting torch.

OXYACETYLENE WELDING

Aluminum. Oxy-Acetylene Welding Sheet Aluminum, J. W. Meadowcroft. Am. Welding Soc.—Jl., vol. 5, no. 6, June 1926, pp. 47–49, 4 figs. Sheet aluminum can be welded with same thoroughness and at much greater speed than sheet metal; rod or strip completely covered with flux or filler body must be used.

Pipe Joints. Design of Oxwelded Equipment, E. E. Thum. Am. Welding Soc.—Jl., vol. 5, no. 5, May 1926, pp. 42–56, 14 figs. Points out that large savings in material can be made when pipe systems are designed for oxyacetylene welded joints; principles which should control design of all oxyacetylene joints.

Precautions and Methods. Do's and Dont's for Oxy-Acetylene Welding, A. Eyles. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 892–894, 1 fig. Requirement for welding; precautions in handling welding equipment; welding sheet metal; flux for aluminum welding; welding galvanized and other coated metals.

P

PATTERNS

Gear-Tooth. Forming Gear Tooth Patterns, A. W. Jansson. Machy. (N. Y.), vol. 32, no. 12, Aug. 1926, p. 939, 1 fig. Describes templet and sanding rolls used in making gear-tooth patterns.

PIGMENTS

Hiding Power. Hiding Power of Pigments, R. L. Hallett. Am. Soc. for Testing Matls.—Preprint, no. 52, for mtg. June 21, 1926, 6 pp., 1 fig. Pigment characteristics which influence hiding power; relation between tinting power and hiding power of pigment in square feet per pound may be calculated if tinting power has been determined.

PIPE

Bends. Bending of a Curved Tube of Circular Cross Section, W. Hovgaard. Nat. Acad. Sci.—Proc., vol. 12, no. 6, June 1926, pp. 365–370, 3 figs. Author worked out solution by new method, and applied resulting formulas to tests on pipe bends carried out at

Mass. Inst. of Technology; deformation of sectional element of curved tube subject to bending.

PIPE, CAST-IRON

Strength. Strength of Iron Pipe Cast by Various Methods, A. N. Talbot. Can. Engr., vols. 50 and 51, nos. 26 and 1, June 29, and July 6, 1926, pp. 699–704 and 713; and 109–115, 11 figs. (Also Contract Rec., vol. 40, no. 36, June 30, 1926, pp. 631–632.) Results of tests by flexure, impact, and internal pressure on 6-in. pipe cast vertically and cast by two different centrifugal processes.

PIPE, STEEL

Electrically Welded. The Mokelumne River Water Supply Project for East Bay Municipal Utility District. West. Constr. News, vol. 1, no. 12, June 25, 1926, pp. 29–35, 15 figs. Successful shop and field tests prove 65 and 63-in. electrically welded steel pipe, 1/2, 7/16, 3/8 in. thick, entirely practicable; details of manufacture and testing.

PIPING

Manufacture and Layout. Pipework: Its Manufacture and Layout, G. H. Willett. Jr. Instn. of Engrs.—Jl. & Rec. of Trans., vol. 36, part 9, June 1926, pp. 380–412, 27 figs. Ferrous pipes; riveted pipes; welded pipes; weldless tubes; fittings; nonferrous pipes; joints; design of layout; velocities; expansion; drainage; steam traps; valves; supports; insulation; corrosion. See also Mech. World, vol. 80, no. 2062, July 9, 1926, pp. 32–33.

PLATES

Ripping Machine. 100-Ton Plate Ripping Machine. Machy. (Lond.), vol. 28, no. 712, June 3, 1926, pp. 245–247, 5 figs. Large machines constructed by Noble & Lund for ripping steel plates to be used in building of Sydney Harbor bridge, and designed for ripping plates from end to end.

PRESSES

Flanging. Special Flanging Press for Bridge Flooring. Engineer, vol. 141, no. 3676, June 25, 1926, pp. 674–675, 4 figs. Multiple hydraulic flanging press for producing steel floor troughings and other special shapes.

Friction. Driving of Friction Presses (Die Antriebsverhältnisse der Friktionspressen), H. Birkle. Praktischer Maschinen-Konstrukteur, vol. 59, no. 17–18, May 1, 1926, pp. 171–174, 3 figs. Examines driving conditions on basis of theories of accelerated rotation and gives calculations for acceleration, power, belting and pulleys, etc.

PRESSWORK

Power. Power Press Work, W. A. Lynch. Machy. (Lond.), vol. 28, no. 712, June 3, 1926, pp. 236–240. Deals with cold sheet-metal work; blanking and piercing; tool clearance, shaving, drawing, forming, stripping, etc.; notes on tool design and press production.

PULLEYS

Loose, Ball-Bearing. Ball-Bearing Loose Pulleys Lower Production Costs. Belting, vol. 28, no. 6, June 1926, pp. 46 and 48. Use of such pulleys in industrial plant in Chicago made it possible to dispense with service of four maintenance men; design of loose pulleys.

Wrought-Iron. Wrought-Iron Pulleys, H. Bentley. Mech. World, vol. 80, no. 2061, July 2, 1926, pp. 6–7, 20 figs. Summarizes advantages claimed for wrought-iron pulleys and discusses some of their features.

PULVERIZED COAL

Boiler Firing. Experiences in Pulverized-Coal Firing (Betriebserfahrungen mit Kohlenstaubfeuerungen), E. Schulz and F. Gropp. Archiv für Wärmewirtschaft, vol. 7, no. 5, May 1926, pp. 121–129, 28 figs. Use of pulverized-coal boilers for peak loads, damages of brickwork in combustion chamber, ash removal, pulverizing plants, evaporating efficiency.

Pulverized-Coal Firing and the Hygroscopic Properties of Lignite. (Kohlenstaubfeuerung und hygroskoische Eigenschaften der Braunkohle), E. Rammler. Archiv für Wärmewirtschaft, vol. 7, no. 6, June 1926, pp. 159–166, 8 figs. Significance of moisture in powdered-coal firing; handling and grinding hygroscopic coals; handling and bunkering of hygroscopic pulverized coal. See also Braunkohle, vol. 25, no. 6, May 8, 1926, pp. 120–131, 8 figs.

Combustion. On Pulverized Combustible (Sur les combustibles pulvérisés), D. W. Lulofs. Chaleur & Industrie, vol. 7, no. 72, Apr. 1926, pp. 177–182, 3 figs. Notes on calculation of cooling effect of furnace walls; data on experiments with furnace having water-cooled top, bottom, and side walls.

Maintenance in Suspension. Power for Maintaining Coal in Suspension, J. G. Coutant. Combustion, vol. 14, no. 6, June 1926, pp. 378–379, 4 figs. Coal pulverized to sufficient fineness will remain in suspension in moving air; this fact forms basis for design of pulverized-coal burners, fuel-transport lines and pneumatic conveying systems; in order to maintain coal in suspension while burning, total air pressure of 1.68 oz. per sq. in. at burner should be provided for.

Utilization. The Use of Pulverized Coal, C. F. Herington. Elec. Light & Power, vol. 4, no. 7, July 1926, pp. 25–28 and 88, 5 figs. Reviews history and describes its use in different plants; practical advantages obtained by this fuel in connection with boiler firing.

PUMPS

Cylinders. Laying Out and Machining Large Pump Cylinders, W. Wheatley. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 908–909, 3 figs. Account of author's experience obtained while operating boring mill in copper-mining center.

Motor-Load Adjustment. Does the Motor Fit the Pump? J. E. Housley. Power, vol. 64, no. 5, Aug. 3, 1926, pp. 158–160, 5 figs. Methods used in testing vacuum pump in order to adjust load on motor and determine capacity of pump.

Power Equipment. Power Drive Equipment for Industrial Pumps, G. Fox. Indus. Engr., vol. 84, no. 7, July 1926, pp. 316–321, 11 figs. Selection of motors, control, and drive equipment for different types of pumps.

Vacuum. Orifice Tests for Rotative Dry Vacuum Pump, R. R. Walden. Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 837–839, 4 figs. Calibrated orifices simplify testing and permit securing pump characteristic curve for checking performance.

PUMPS, CENTRIFUGAL

Automatic Control. Controlling Centrifugal Pumps Automatically, A. H. Hubbell. Eng. & Min. Jl.-Press, vol. 121, no. 25, June 19, 1926, pp. 997–1001, 9 figs. Manual attendance largely eliminated; labor cost lessened; system is dependable and applicable to practically every condition of mine drainage where water is handled by units of this type.

R

RAILLESS TRACTION

Ipswich, England. Successful Railless Enterprise. Motor Transport, vol. 43, no. 1112, July 5, 1926, pp. 21–22, 2 figs. Some facts and figures explaining why trackless trolley cars are favored by Ipswich Corporation.

RAILS

Breaking of. Accidental Breaking of Rails (Les ruptures accidentelles des rails), M. Merklen. Revue Générale des Chemins de Fer, vol. 45, no. 5, May 1926, pp. 346–387, 66 figs. Defects in rails and their remedies; brittleness, cracks and fissures and their spread in rails; pipe, segregation, premature wear, corrosion, and wear.

Initial Causes of Rail Breakages, and Methods Employed to Reduce the Number, L. Sister. Int. Ry. Congress—Bul., vol. 8, no. 6, June 1926, pp. 502–506, 8 figs. Causes and methods of reducing breakages; improving metal in ingots by localizing piping in head by keeping it heated.

RAILWAY MANAGEMENT

Stores Control. Standardization and Simplification of Stores Stock. Ry. Age, vol. 80, no. 30, June 11, 1926, pp. 1635–1638. Report of committee before Am. Ry. Assn.

RAILWAY MOTOR CARS

Benzol. New Benzol Car of the Town of Spremberg [Ein neuer Benzoltriebwagen der Stadt Spremberg (Lausitz)], Fratschner. Verkehrstechnik, vol. 39, no. 18, Apr. 30, 1926, pp. 288–291, 3 figs. Design and operation of A.E.G.-car replacing steam service and considerably reducing operating costs.

Gasoline. C. B. & Q. Acquires Gasoline Car with Independent Power Units. Ry. Age, vol. 81, no. 1, July 3, 1926, pp. 19–20, 3 figs. New car having seating capacity for 42 passengers and containing baggage and mail compartments; entire floor space of 65 ft. available for revenue purposes; trucks provided with 100-hp. motors.

Trucks. A New Truck System for Tramway Cars and Rail Cars. Tramway & Ry. World, vol. 59, no. 28, June 17, 1926, pp. 281–286, 10 figs. New type of truck with Cardan drive adapted to meet requirements of railroads with light motor coaches.

RAILWAY OPERATION

Train Control. New York Central Equips Engines for Train Stop. Ry. Signaling, vol. 19, no. 7, July 1926, pp. 264–266, 9 figs. Locomotive taken from service for 33 hours is equipped in average of 93.7 man-hours; installation of wayside inductors.

Train Control Inspection Equipment. Ry. Signaling, vol. 19, no. 7, July 1926, pp. 280–281, 4 figs. Electrical tests carried out at engine terminals in maintenance program of Richmond, Fredericksburg & Potomac R. R.

Train Despatching. Time Schedules and Dispatching Methods, J. A. Droege. Ry. Rev., vol. 79, no. 1, July 3, 1926, pp. 17–18. Positive meet system possesses advantage of simplicity over superiority by direction.

RAILWAY REPAIR SHOPS

Car. The Lackawanna Has an Unusual Car Shop. Ry. Rev., vol. 78, no. 24, June 12, 1926, pp. 1067–1070, 11 figs. In Keyser Valley shops at Scranton, Pa., average of 450 freight cars are rebuilt each month with force of 430 men.

RAILWAY SHOPS

Materials Handling. Handling Material in Railroad Shops. Ry. Mech. Engr., vol. 100, no. 7, July 1926, pp. 447–454, 25 figs. Developments in equipment have facilitated work of routing material; efficient operation depends on prompt deliveries.

Pacific Coast. Pacific Coast Railroad-Shop Methods, F. A. Stanley. Am. Mach., vol. 64, no. 25, June 24, 1926, pp. 971–972, 5 figs. Practice in Sacramento shops of Southern Pacific Co.

RAILWAY SIGNALING

Maintenance. Signal Maintenance on the Erie. Ry. Signaling, vol. 19, no. 7, July 1926, pp. 257–263, 17 figs. Performance in terms of train interruption improved 76 per cent and expenditure for battery reduced 43 per cent during last seven years.

RAILWAY SWITCHES

Renewal. How to Renew Slip Switches, J. Wilson. Ry. Rev., vol. 79, no. 2, July 10, 1926, pp. 45–46, 2 figs. Two methods for renewing slip switches; compares their advantages and stresses importance of good maintenance.

RAILWAY TRACK

Curves. Influence of Gyroscopic Action on the Necessary Superelevation to Be Provided on Curves, F. Corini. Int. Ry. Congress—Bul., vol. 8, no. 6, June 1926, pp. 491–501, 7 figs. Gyroscopic action as affecting railway rolling stock and passing round curve; superelevation of outer rail; superelevation of outer rail to gyroscopic moment, etc.

Drainage. Illinois Central Installs Large Mileage of Track Drainage, P. T. Savage. Ry. Eng. & Maintenance, vol. 22, no. 7, July 1926, pp. 260–262, 3 figs. Improper drainage of subgrade, permitting formation of soft spots and water pockets in roadbed, has given maintenance force constant trouble; combination vitrified pipe and field tile drain proves efficient and economical.

Track Stresses. Report of the Bridge Sub-Committee on Track Stresses, 1925. India Govt. Publication—Tech. Paper, no. 245, 1926, 54 pp., 83 figs. Sources of information; note on elastic theory; general conclusions regarding locomotive, cars, particular types of locomotives, curved track, effect of sleeper spacing, etc.

REFRACTORIES

Foundry. Refractories in the Foundry, H. V. Grundy and A. Phillips. Foundry Trade Jl., vol. 33, nos. 511 and 513, June 3 and 17, 1926, pp. 387–391 and 445–448, 15 figs. Chemical and physical properties of molding sands; test for cohesiveness; durability sand; ground coke, ground gas carbon and ground charcoal. June 17: Refractories for cupola; lining corrosion; refractory coatings.

High Temperatures and. Refractories and High Temperatures. Eng. & Boiler House Rev., vol. 39, no. 11, May 1926, pp. 515–516, 1 fig. Discusses complicated problems with high-grade firebricks.

Specific-Heat Determination. The Role of Specific Heat in the Selection of Refractories, A. E. MacGee. Am. Ceramic Soc.—Jl., vol. 9, no. 6, June 1926, pp. 374–379. Discussion of specific heats in which particular emphasis is placed upon its determination at high temperatures; methods used in past for its determination and application of values in selection of refractories; points out need for determining physical properties of brick which have been in service.

Steel and Metal Industries. Refractory Materials for Steel and Metalworking Industry (Feuerfeste Stoffe für die Eisen und Metall erzeugende Industrie), E. H. Schulz. Zeit. des Vereines deutscher Ingenieure, vol. 70, no. 12, Mar. 20, 1926, pp. 408–410. Methods of investigating, testing and standardization of refractories for metallurgical industry; main types of refractory brick and their properties; difficulties of selection and testing.

REFRIGERATING MACHINES

Absorption. Absorption Refrigerator Without Moving Parts. Engineering, vol. 122, no. 3157, July 16, 1926, pp. 75–76, 6 figs., partly on p. 74. Type of small self-contained refrigerating plant developed by B. de Platen and C. G. Munters, Stockholm, using ammonia as refrigerating medium in which entire plant is hermetically sealed.

Domestic. Self-Contained Unit or Separate Systems, O. T. Knight. Am. Gas Jl., vol. 125, no. 2, July 10, 1926, pp. 32 and 38. Factors to be considered in choosing refrigerating system for domestic purpose; objections to self-contained unit; brine-circulating systems and basement installation.

Leipzig Fair, Germany. Ice and Refrigerating Plants at the Technical Spring Fair in Leipzig (Eismaschinen und Kuhlanlagen auf der diesjährigen Technischen Frühjahrsmesse in Leipzig), Wärme- u. Kälte-Technik, vol. 28, no. 9, May 5, 1926, pp. 97–99, 8 figs. Various types of small machines used for food industry, freight, and domestic purposes, including those of Lanz, Esslingen, Brown Boveri, Rumpler, Gebr. Bayer.

REFRIGERATING PLANTS

Reconstruction. Reconstructing a Refrigerating Plant, C. L. Morgan. Power, vol. 64, no. 5, Aug. 3, 1926, pp. 161–162. Problems encountered in enlarging hotel refrigerating system; old stack too small for natural draft; new two-stage compressor installed.

ROLLING MILLS

Bar Mills. Merchant Bar Mills, J. D. Knox. Iron Trade Rev., vol. 78, no. 25, June 24, 1926, pp. 1617–1621, 7 figs. Describes rebuilding program at Cambria Works of Bethlehem Steel Co.; modern units with facilities for handling, shipping, and warehousing products replace high-cost units.

Blooming Mills. Blooming Mill of Massive Size. Iron Age, vol. 118, no. 6, Aug. 5, 1926, pp. 348–350, 4 figs. Largest and heaviest blooming mill in world has been completed at Youngstown works of United Engineering & Foundry Co., for Homestead works of Carnegie Steel Co.; this is 54-in., 2-high, reversing blooming mill.

Comparative Investigations of Power Requirements and Rolling Efficiency of Blooming Mills (Vergleichende Untersuchungen auf Kraftbedarf und Walzleistung an Blockstrassen), C. Schmitz. Stahl u. Eisen, vol. 46, no. 23, June 10, 1926, pp. 769–776, 12 figs. Investigations, carried out with aid of operating data and time studies, show what conditions influence power consumption and efficiency; calcula-

tion also provides operator with adequate means of supervising blooming mill.

Electric Drive. Converting Existing Rolling Mills from Steam-Engine to Electric-Motor Drive, D. M. Petty. Elec. Jl., vol. 23, no. 6, June 1926, pp. 276–278, 4 figs. Cites instances in which number of causes led to replacement of steam engine by motor; one of principal advantages of electric power over steam is difference between no-load losses; other advantages.

Reversing Mill Driving Equipments, H. E. Stokes. Elec. Jl., vol. 23, no. 6, June 1926, pp. 306–310, 10 figs. Advantages of d.c. motor drive; reversing-mill equipment built by Westinghouse Co.

Hot-Strip Mills. New Hot-Strip Mill in Operation, R. A. Fiske. Iron Age, vol. 118, no. 2, July 8, 1926, pp. 78–81, 6 figs. New 14-stand hot-strip mill of Acme Steel Co., Riverdale, Ill.; stands are driven by 9 motors; 101 motors on runout table and hot bed; producer gas used.

Plate Mills. Steel Works Modernisation. Elec. Rev., vol. 98, no. 2534, June 18, 1926, pp. 902–905, 7 figs. Electrical equipment of new plate mill of Consett Iron Co., Ltd., England.

ROLLS

Cast-Iron. Making Rolls From Semi-Steel. Foundry Trade Jl., vol. 33, no. 514, June 24, 1926, p. 474. Describes method of manufacturing cast-iron rolls from semi-steel which led to conclusion that mixture of liquid steel and once-melted iron of suitable composition will give good reliable results at reasonable cost.

S

SAND BLAST

Air Ratio. Compressed Air and Sand Blast Operation (Die Pressluftverhältnisse beim Betrieb von Sandstrahlgebläsen), R. Karg. Zeit. für komprimierte und flussige Gase, vol. 25, nos. 2 and 3, Feb. and Mar. 1926, pp. 17–20 and 30–33, 5 figs. Discusses quantity of air required and its calculation from number and diameter of nozzles, operating pressure, temperature and type of compression.

SAND, MOLDING

Gas Permeability and Cohesion. The Practical Utilization of Apparatus for Measuring the Permeability and Cohesion of Moulding Sands, R. Lemoine. Foundry Trade Jl., vol. 33, no. 513, June 17, 1926, pp. 449–452, 3 figs. Apparatus for measuring permeability and cohesion in cold state, when employed according to certain methods, may be of great service in foundry; controlling mixtures, raw materials, and prepared sand.

Heat Transmission. Heat-Transmission Tests on Molding Sands (Wärmeübergangsversuche an Formsanden), F. Roll. Giesserei-Zeitung, vol. 23, no. 11, June 1, 1926, pp. 298–301, 9 figs. Theory of heat transmission; heat conduction, convection and radiation; heating curves and results of tests.

Reclamation. Reclaims Foundry Sand, A. A. Grubb. Foundry, vol. 54, no. 12, June 15, 1926, pp. 451–463. Results of sand reclamation at Ohio Brass Co., Mansfield, O.; new sand cost cut from $1.00 per ton of castings to 20 cts. in three years; during same period loss is reduced from 8.4 to 4 per cent.

SAWS

Circular. Reversible Single Belt Circular Saw and Moulding Machine, A. Bahis. Eng. Progress, vol. 7, no. 6, June 1926, pp. 156–157, 3 figs. Describes combined or universal woodworking machine, suitable for smaller plants.

SCREW THREADS

Calipers. Screw Thread Mircometer Calipers, H. Bentley. Indus. Mgmt. (Lond.), vol. 13, no. 6, June 1926, pp. 261–262, 2 figs. Application of micrometer calipers to measuring V-type screw threads.

Generating Machine. Thread Generating and Forming Machine. Machy. (Lond.), vol. 28, no. 716, June 30, 1926, pp. 373–374, 3 figs. Cornelis machine, which is semi-automatic in operation and consists of lathe type bed with fast headstock at one end and saddle which is guided along bed by two cylindrical steel bars supported between headstock and bracket, at left-hand end of bed.

Measuring Instruments. Universal Measuring Microscope (Ein Universalmessmikroskop), A. Steinle. Maschinenbau, vol. 5, no. 10, May 20, 1926, pp. 445–449, 15 figs. Discusses optical measurement of thread and progress attained in increased precision with projection method.

SEAPLANES

Besson M.B.-35. The Besson M.B.-35 Seaplane. (L'hydravion Besson M.B.-35), Aéronautique, vol. 8, no. 85, June 1926, pp. 211–213, 5 figs. 120-hp. monoplane capable of ascending 6500 ft. in 13 minutes with useful load of 500 lb.; can be dismantled and transported in tube 6½ ft. in diameter by 23 ft. long.

Hull Designing. Designing Hulls of Seaplanes, (Le tracé des carènes d'hydravions), Benoit. Aéronautique, vol. 8, no. 85, June 1926, pp. 199–207, 15 figs. Marine considerations; general properties of pontoons; determining form of keel, chine, and transverse sections; transverse stability, fins, etc.

SHAFTS

Rubbing. Shaft Rubbing, B. L. Newkirk. Mech. Eng., vol. 48, no. 8, Aug. 1926, pp. 830–832, 6 figs. Relative freedom of rotor shafts from sensitiveness to rubbing contact when running above their critical speeds.

SHAPERS

Portable Draw-Cut. A British Portable Draw-Cut Shaper. Brit. Machine Tool Eng., vol. 3, no. 39, May-June 1926, pp. 425–427 and 430, 2 figs. Butler portable 48-in. draw-cut shaper designed for machining operations on heavy units such as engine beds or housing frames of mill rolls; ram movements cover 9 ft. horizontally and 6 ft. vertically.

SHEARS

8-in. Billets, for. Shearing Machine for 8 in. Billets. Engineer, vol. 142, no. 3681, July 30, 1926, p. 125, 1 fig. Machine exhibited by H. Pels & Co., at Spring Fair at Leipzig; machine is of steel throughout; main drive is from 100-hp. motor mounted on staging at top.

Upcut. Upcut Shear Affords Even Edges. Iron Trade Rev., vol. 79, no. 5, July 29, 1926, pp. 255–257, 5 figs. Unit equipped with knives which can be raised or lowered while oscillating; design makes possible use of stationary table; description of shearing cycle.

SHEET METAL

Shell Blowing. "Blowing" a Shell of Heavy Sheet Steel, E. Sheldon. Am. Mach., vol. 65, no. 6, Aug. 5, 1926, pp. 233–234, 3 figs. Making shell to replace casting; tallow is used as expanding medium; tools are quite simple.

SILK

Artificial. The Advance of Rayon, O. Wilson. Indus. & Eng. Chem., vol. 18, no. 8, Aug. 1926, pp. 829–831, 2 figs. Discusses four distinct processes for commercial production of artificial fibers; all of them start with cellulose and end up with yarn of fine luster, but chemical and mechanical steps in between, and qualities of resulting product, are different.

SMOKE

Abatement. Smoke Elimination. Clay-Worker, vol. 85, no. 6, June 1926, pp. 463–465, 5 figs. Process developed by W. C. Mitchell of Superior Press Brick Co., St. Louis, by which smoke menace is eliminated in plant manufacturing brick and tile.

The Black Smoke Problem: The New Smoke Abatement Bill, J. B. C. Kershaw. World Power, vol. 5, no. 30, June 1926, pp. 310–314. Present law relating to smoke emission; report of Committee on Smoke Abatement; set up in 1914; Government bills of 1923 and 1925; probable effects of new legislation.

SPRINGS

Elliptic, Chart for. Elliptic Spring Chart, E. Latshaw. Machy. (N. Y.), vol. 32, no. 11, July 1926, pp. 888–889, 1 fig. Presents chart for use in designing elliptic springs.

Design of Helical Springs for Compression or Extension. Am. Mach., vol. 65, no. 5, July 29, 1926, p. 215, 2 figs. Formulas for computing helical springs. Reference-book sheet.

Helical. Helical Springs, H. Sabine. Machy. (Lond.), vol. 28, no. 710, May 20, 1926, pp. 177–178, 2 figs. Tests carried out on certain highly loaded springs used in various apparatus designed by author.

Laminated. Experiments on Laminated Springs, H. S. Rowell. Automobile Engr., vol. 16, no. 216, June 1926, pp. 229–236, 44 figs. Experimental determination of spring stresses and observations on spring friction.

Mechanical. The Specification and Control of Mechanical Springs, J. K. Wood. Mech. Eng., vol. 48, no. 8, Aug. 1926, pp. 808–814, 6 figs. Gives list of available specifications for heavy springs and tabulates features of these specifications; introduces idea of load-deflection sector to be used in place of load-deflection curve and employs idea in framing sample specifications.

Standardization. Standardization of Springs from the Standpoint of Construction (Ueber die Normung von Federn vom Standpunkt der Konstruktion), R. Geyer. Maschinenbau, vol. 5, no. 10, May 20, 1926, pp. 450–451, 8 figs. Discusses possibility of standardizing pressure and expansion springs and proposes standard dimensions based on calculations.

STANDARDIZATION

Developments. Engineering and Industrial Standardization. Mech. Eng., vol. 48, no. 7, July 1926, pp. 763–764. Proposals for standardized Woodruff keys; T-slot and parts; standards for gear-tooth form and nomenclature; for plain taper and gib-head taper keys; bolt, nut and rivet proportions; standardization of machine tapers; lathe and planer tool holders; standards for plain and lock washers.

Size Determination. Examples of Size Determination, E. R. Hedrick. Mech. Eng., vol. 48, no. 8, Aug. 1926, pp. 842–844, 3 figs. Presents examples of size determinations on two widely different products: heavy safes and cranes.

STANDARDS

German N.D.I. Reports. Report of German Industrial Standards Committee (N.D.I.-Mitteilungen), W. Reichardt. Maschinenbau, vol. 5, no. 10, May 20, 1926, pp. 481–483. Proposed standards for square nuts and holes; bronzes and red brass, terminology, castings, etc.; diameters for woodworking machines; milling cutters and arbors.

Report of German Industrial Standards Committee (N.D.I. Mitteilungen). Maschinenbau, vol. 5, no. 11, June 3, 1926, pp. 533–544, 18 figs. Proposed standards for headless screws with square shoulder; gears, definitions, designations and graphic symbols; welding, definitions and graphic symbols.

STEAM

High-Pressure. Industrial Production of High-Pressure Steam (La production industrielle de la vapeur d'eau à haute pression), M. C. Roszak and M. Veron. Société des Ingenieurs Civils de France, no. 8, Apr. 30, 1926, pp. 159–168. Discusses influence of high pressure on utilization and production of steam, concludes that high-pressure steam leads to important improvements in efficiency of central stations; pressures of 50 kg. per sq. cm. and of 80 kg. per sq. cm. will have to be considered.

Pressure Regulation. A New Pressure-Reducing Apparatus (Ein nieuw system reduceerapparaat), A. Bargeboer. Ingenieur, vol. 41, no. 17, Apr. 24, 1926, pp. 333–337, 4 figs. Dikkers-Bargeboer system of pressure regulation, calculation and applications.

Tables. New Tables of the Properties of Steam. (Nouvelles Tables des Propriétés de la Vapeur d'eau), R. Martin. Chaleur & Industrie, vol. 7, no. 73, May 1926, pp. 269–271. Extensive metric tables and charts of properties up to a pressure of 60 atmospheres, values below 30 atmospheres corresponding to those established by Knoblauch, Raisch, and Hausen, and those above to values of Dr. R. Mollier.

STEAM ENGINES

M. E. P. Indicator. The Direct Indication of Mean Effective Pressure—The Fi Meter [Das direkte Anzeigen des mittleren indizierten Kolbendruckes (das Fi-Meter)], J. Geiger. Wärme, vol. 49, no. 19, May 7, 1926, pp. 331–333, 10 figs. Describes apparatus combining pressure-volume and pressure-time measurements and giving direct reading or recording of mean effective pressure.

Reciprocating. Reciprocating Engine Uses Steam at 686 Degrees. Power, vol. 63, no. 24, June 15, 1926, pp. 918–921, 6 figs. Cross-compound non-condensing poppet-valve-Corliss combination, designed for 350-lb. pressure and 250-deg. superheat, at plant of Dow Chemical Company, Midland, Mich.

Uniflow. An Auxiliary Steam Engine Development. Mar. Engr. & Motorship Bldr., vol. 49, no. 586, June 1926, pp. 216–219, 8 figs. Economy to be derived from use of uniflow principles for marine auxiliary-engine purposes.

Record Breaking Uniflow Engine in Steel Mill. Power Plant Eng., vol. 30, no. 13, July 1, 1926, pp. 746–750, 7 figs. Largest uniflow in America, incorporating many unusual features of design and construction, is pioneer unit for reversing blooming-mill service at Steubenville plant of Wheeling Steel Corp.

STEAM PIPES

High-Pressure. High-Pressure Pipe Lines for Highly Superheated Steam of 40 Atmos. (Hochdruck-Rohrleitungen für hoch überhitzten Dampf von 40 atü), H. Menk. Archiv. für Wärmewirtschaft, vol. 7, no. 5, May 1926, pp. 133–138, 22 figs. Discusses question of materials, calculation of thickness of pipe, fixing of flanges by DIN standards, expansion pipes and their tests, valves and fittings, draining of pipes, etc.

Power-Plant. How to Lay Out Power-Plant Piping, S. Crocker. Power, vol. 63, nos. 20, 23, and 25, May 18, June 8, and 22, 1926, pp. 769–770, 890–891 and 962–965, 8 figs. May 18: Preparation of preliminary plans. June 8: How to select pipe sizes. June 22: Influence of high temperatures on choice of materials; development of dimensional standards.

STEAM POWER PLANTS

Cold Circulating Water. The Value of Cold Circulating Water. Power Engr., vol. 21, no. 243, June 1926, pp. 220–221. Investigation into reduction of operating costs of steam power plants possible by use of lower-temperature circulating water.

Control Apparatus. Modern Devices for Power Plant Control. Chem. Age, vol. 14, no. 365, June 26, 1926, pp. 567–569, 2 figs. Notes on various types of apparatus applicable to conservation of power in works and for obtaining accurate data for costing.

Diphenyl Oxide Bi-Fluid. Diphenyl Oxide Bi-Fluid Power Plants, H. H. Dow. Mech. Eng., vol. 48, no. 8, Aug. 1926, pp. 815–818, 5 figs. Author proposes plant working between 750 and 70 deg. fahr.; diphenyl-oxide vapor would be generated in boiler at 135 lb. gage pressure and expanded to vacuum depending on steam pressure with which it is desired to work; temperature-entropy diagrams show two such combinations, one in which diphenyl oxide is condensed at atmospheric pressure, making steam of about 600 lb. pressure, and other exhausting at about 24 in. of vacuum, making steam of 150 lb. pressure.

High-Pressure. Tests of a 60-Atmosphere-Pressure Steam Plant at the Borsig Works. Mech. Eng., vol. 48, no. 8, Aug. 1926, pp. 854–855, 2 figs. Review of work by E. Josse, published in Zeit. des. Vereines deutscher Ingenieure, May 22, 1926.

Laundry. Independent Towel Supply Co.'s New Power Plant. Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 828–833, 7 figs. Non-condensing 500-kw. turbo-generator, coal and ash-handling equipment, combustion-control system and laundry water heaters form points of interest.

Stairs and Ladders. Safety Demands Proper Stairs and Ladders. Power Plant Eng., vol. 30, no. 15, Aug. 1, 1926, pp. 855–857, 4 figs. Design, construction and use of proper facilities for climbing and working at elevations are notable factors in preventing accidents.

Textile Mills. Balfour Mill Insures Operation by Installing Power Plant. Power, vol. 64, no. 5, Aug. 3, 1926, pp. 154–157, 8 figs. Installation of power plant by Balfour Mill located at Smyth, N. C.; equipped with 1000-kw. turbo-generator unit and single-retort side-dump underfeed stokers.

STEAM TURBINES

Automatic Regulation. Automatic Regulation Maintains Exhaust and Bleeder Pressures. Power, vol. 63, no. 26, June 29, 1926, pp. 1005–1007, 4 figs. Describes 500-kw. Moore turbine at power plant of Lever Bros. Co. at Cambridge, Mass.; although there are three governing devices, operation is simpler than